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WELDING ELECTRODES.

BY HUGO W. HIEMKE.*

During the last decade the use of electric arc welding for fabrication of many forms of engineering structures has increased tremendously. This increased use of welding has been augmented by the development of covered welding electrodes which are capable of depositing weld metal as strong and ductile as the metals to be joined.

The purpose of this article is to tabulate available information, and submit a certain amount of new test information. Also, the author classifies the various service requirements for welding electrodes.

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History of Electrode Developments.

Electric welding originated in some experiments by a Russian scientist named Bernados, who, in 1887, demonstrated the feasibility of locally heating heavy masses of metal with a carbon arc. Shortly thereafter one of his countrymen, Slawianoff, made the first use of a metal electrode for effecting such localized heating. For a long time the practical application of these early experiments was hindered by lack of knowledge concerning the proper compositions to be used for welding electrodes. By the turn of the century sufficient experimental work had been done in Russia, Germany, and Sweden to determine that a pure iron (commonly known in this country as Swedish iron) produced the best results in electric welding of steel. The best results of the early investigators when compared with the modern art were indeed mediocre. However, they represented an important step in the development of the art. The first efforts to improve ordinary bare wire electric arc welding were directed toward the stabilization of the arc. It was found that the ordinary lime wash used in drawing of wire produced a wire which welded more satisfactorily than one which was completely clean of lime. This undoubtedly led investigators to a study of materials other than lime for stabilizing purposes. Titanium oxide and mill scale and combinations of the two were frequently used in the early days as arc stabilizers.

Further studies concerning the effect on the arc of various materials applied to electrodes in the form of coatings indicated that with the proper combinations of chemicals improved physical properties of deposited metal could also be obtained. Experiments with much heavier coatings or coverings by German, English, and Swedish investigators between 1907 and 1914 did much to develop the present day heavily coated or covered electrode. These coverings consisted of slag forming ingredients such as asbestos or calcium silicates of the acid open hearth slag type.

During the period from 1914 to 1918 the use of electric welding was stimulated greatly by the demands for increased manufacturing rates for production of war materials. One of the most significant developments of this period was that of the cellulosic type covered electrode. In its earliest form this covering consisted

simply of a heavy wrapping of paper impregnated with sodium silicate and baked to drive off excess moisture. During the 1920's further experiments with the mineral type of electrode covering produced weld metal properties approaching the excellence attained by the use of the present day electrodes. At the same time electrode coatings were evolved which used a combination of cellulosic materials with slag forming ingredients. The term " shielded arc welding " has been applied to welding which is performed by the use of covered electrodes of the cellulosic, mineral or composite types. It was the advent of the shielded arc electrode which stimulated the use of electric arc welding for important construction uses. Accordingly, it appears desirable to discuss in some detail the differences between the old bare wire and the more modern shielded arc types of electrodes.

Physical Differences Between Bare and Shielded Arcs.

One of the most obvious differences between arcs struck with bare wire electrodes and shielded arc electrodes is the great difference in voltage for the same arc length. The voltage across an electric arc depends on the length of the arc and the arc atmosphere. In the case of bare wire, this atmosphere is practically pure air, although a slight contamination by the gases evolved from the melting metal may occur. In the case of shielded arcs, the arc atmosphere is a function of the electrode covering. When cellulosic type coverings are used, the burning of the electrode is accompanied by the evolution of large volumes of hydrogen and carbon monoxide gases. The arc space also has ionized particles of the vapors of certain of the slag forming constituents and metals. A study of the effect of various pure gases on the voltage of a one-eighth inch long electric arc, using iron electrodes, has shown that the voltage in air is 22 volts; in hydrogen, it is 40 volts; in carbon monoxide, 30 volts, and in nitrogen, 28 volts. It is apparent at once that arc atmospheres containing appreciable quantities of hydrogen may be expected to require a higher voltage for the maintenance of an arc during welding. This is confirmed by the fact that the cellulosic type of covering operates with a higher voltage than the mineral type of covering. Both types of shielded arc electrodes, however, operate at a considerably higher voltage than a

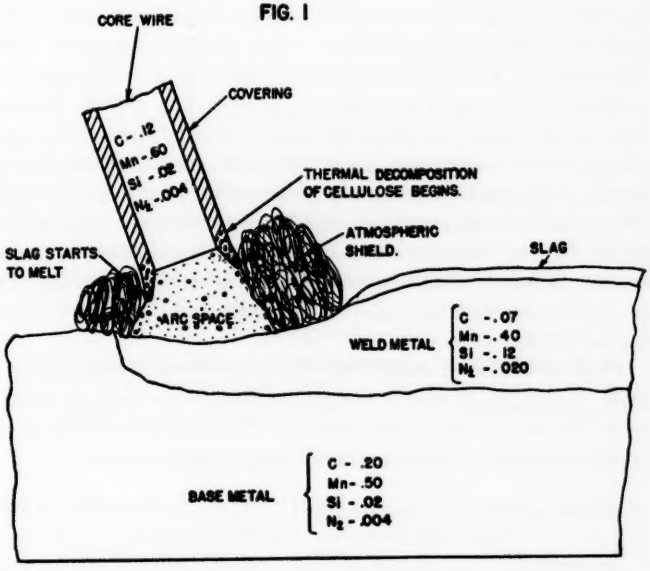
bare wire arc. From a practical standpoint, this increased arc voltage means increased arc energy which, in turn, results in greater penetration into the base metal during welding.

Another difference which is noted in the operation of shielded arc electrodes as compared with bare wire electrodes is the fact that bare wire electrodes operate most satisfactorily with the electrode attached to the negative terminal of the welding generator. In the case of shielded arc electrodes of the cellulosic type, the metal deposition is most effective when the polarity is reversed, that is, the electrode is attached to the positive terminal of the welding generator. Certain combinations of mineral coverings may be used with either electrode negative or electrode positive. In such cases it has been found that alternating current may also be used quite successfully. In general, it may be stated that alternating current should not be used for the deposition of bare wire or shielded arc electrodes of the cellulosic type or other shielded arc electrodes, which operate most satisfactorily with electrode positive.

One of the most significant differences between bare wire welding and shielded arc welding is in the mechanism of metal transfer. Ordinary observation of the welding arc through a face shield does not give reliable testimony to the striking difference which exists. In the case of bare wire arcs, the deposition is accomplished by the melting of a globule, necking down of the globule by surface tension, and transfer to the base metal either by gravitational forces or in the case of overhead welding by a combination of magnetic and surface tension effects. Shielded arc deposition is accompanied by the transfer of large globules of slag and metal at a considerably decreased periodicity than is encountered in bare wire arc welding. However, these globules do not account for the major portion of arc metal transfer. Larson (1) has shown, by means of experiments in which metal probes were passed through the arc, that the major portion of the metal is transferred in the form of very tiny droplets. Figure 1 illustrates graphically the size and distribution of particles in a shielded arc as visualized from the experiments conducted with metal probes. In general, the particles ranged in size from .001 inch in diameter to .010 inch in diameter, and there were approximately 100 particles of the smaller size compared with one

particle of the larger size. Other experiments conducted in this investigation indicated that the particle velocity was of the order of nine feet per second and that the total volume of the particles in the arc space at any one time did not exceed one-tenth of one per cent. This latter demonstration shows quite clearly why it is impossible to visualize how these particles go through the arc without the aid of rather specialized tests.

FIG. 1



SKETCH SHOWING METAL TRANSFER IN SHIELDED ARC WELDING PROCESS.

Figure 1 also shows a cross section of a shielded arc electrode and the deposited metal. This diagram is intended to illustrate the important physical and chemical changes which take place during the deposition of a bead of weld metal. It will be noted that the electrode covering extends beyond the end of the electrode a sufficient amount to produce a sort of crater or crucible from which the metal particles are expelled. The burning of the electrode covering

produces the arc atmosphere which in the case of cellulosic type coverings is largely hydrogen and carbon monoxide gas. In the case of mineral covered electrodes, this atmosphere contains, in addition to smaller volumes of hydrogen and carbon monoxide, the vapors of the non-metallic and metallic ingredients of the covering. The effect of this gas evolved from the covering is to expel the air, permitting the metal to be transferred from the electrode tip to the pool of weld metal with a minimum of contamination. At the same time the slag forming ingredients of the covering melt and form a layer of molten slag which floats on top of the weld metal. This slag layer has a number of functions, the principal ones being the protection of the weld metal during solidification and the retardation of the cooling of the weld metal. During the solidification period, which is extremely short but nevertheless sufficient for an approach to equilibrium chemistry, at the very high temperatures of the molten metal, reactions take place between the slag and the metal. There is evidence to show that the deposited metal has an entirely different chemistry from the metal in transit through the arc space; notably a reduction in manganese and silicon occurs during solidification.

From an economic standpoint, the use of covered electrodes has made it possible to employ larger electrode sizes than are practicable for use in bare wire welding. This results in greatly increased welding speeds and reduced welding costs.

Chemical and Metallurgical Differences Between Bare Wire and Shielded Arc Weld Metal.

Probably the most significant difference between bare wire deposited weld metal and shielded arc weld metal concerns the ductility of the weld metal. As a matter of fact the developments in electrode coatings have been directed largely toward the improvement of the property of ductility to a point where the deposited weld metal is approximately of the same ductility as the metals used in construction. Accompanying this improved ductility there have been significant chemical and metallurgical changes in the weld metal. Table 1 illustrates typical chemical analyses of bare wire, lightly coated, and shielded arc weld metals. It will be noted that

TABLE I

CHEMICAL ANALYSES OF WELD METALS.

	Bare	Lightly Coated	Covered (Shielded Arc)
Carbon	.05	.06	.08
Manganese	.10	.20	.50
Silicon	.01	.05	.15
Phosphorus	.02	.02	.02
Sulphur	.03	.03	.03

the shielded arc metal contains more carbon, manganese, and silicon than the bare wire weld metal, while the phosphorus and sulphur analyses remain practically the same. The chemistry of the weld metal deposited by lightly coated electrodes is intermediate between the other two. Table 2 illustrates the differences in oxygen and

TABLE – 2

FRACTIONAL OXYGEN DETERMINATION OF WELD METALS (HOYT).

	Bare	Shielded Arc
Oxygen as Fe O	.282	none
Oxygen as Mn O	none	.010
Oxygen as Si O_2	.0004	.052
Oxygen as Al_2 O_3	.016	.008
Total Oxygen	.298	.070
Total Nitrogen	.150	.010

nitrogen content of bare and shielded arc weld metals. These analyses, although not ordinarily made, illustrate very effectively the important function of a shielded arc covering in expelling the air from the arc space. The high oxygen and nitrogen content of bare

wire weld metal is primarly responsible for the low ductility of this weld metal.

TABLE - 3

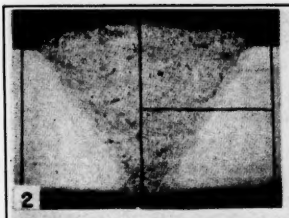
PHYSICAL PROPERTIES OF BARE AND SHIELDED ARC WELD METALS.

	Bare	Shielded Arc
Yield Point	35,40,000 psi	50-55,000 psi
Tensile Strength	50-60,000 psi	60-65,000 psi
Elongation in 2 in.	6-12%	18-35%
Reduction of Area	8-20%	30-60%
Charpy Value	2 ft. lbs.	20-40 ft. lbs.
Brinell Hardness No.	100-110	120-130

Table 3 shows the comparison of physical properties of bare wire and shielded arc weld metal. It will be noted that the shielded arc weld metal is somewhat stronger than bare wire weld metal, but the most significant difference is in the elongation values commonly measured to evaluate the property of ductility. It will also be noted that the Charpy impact value of shielded arc metal is much higher than the Charpy value of bare wire weld metal.

Figures 2 to 5 inclusive illustrate the differences in micro-structure between these two types of weld metal. Figures 2 and 3 are micro-sections of multilayer welds made with bare and shielded arc electrodes. In the case of the bare wire weld metal numerous large inclusions are present. The shielded arc weld metal is substantially free of inclusions. It will be noted that the main body of the weld metal in case of shielded arc electrodes is fully recrystallized by the heat of the superimposed layer of weld metal. In the case of bare wire welding, such complete recrystallization is not feasible.

Figures 4 and 5 illustrate the structure of bare and shielded arc weld metal at 500 diameters after heat treating and etching to determine the presence of nitride needles. It will be noted that a large number of nitride needles are present in the bare wire weld metal, while these needles are absent in the shielded arc weld metal.



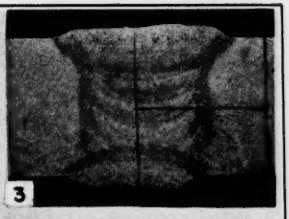


FIG. 2 CROSS-SECTION OF BARE WIRE WELD-ETCHED.

FIG. 3 CROSS-SECTION OF COVERED ELECTRODE WELD-ETCHED.

FIG. 4 MICROSTRUCTURE OF BARE WIRE WELD METAL X 500.

FIG. 5 MICROSTRUCTURE OF COVERED ELECTRODE WELD METAL X 500.

Considerable emphasis has been placed on the property of ductility in connection with the improvement of welding electrodes. Although ductility is not directly useful in the design of structures, it exerts an important influence on the serviceability of structures and also has an influence on the ease of fabrication. During the welding operation, particularly when welding is performed on rigid structures, the weld metal itself is called upon to yield plastically during the operation of cooling. Weld metal of low ductility would be incapable of absorbing these plastic movements under the fixed conditions which exist, particularly in heavy work. Quite frequently it is convenient to perform cold forming operations on structures after welding. In order to withstand such operations, a weld metal of high ductility is absolutely necessary.

In addition to the normal service stresses which a designer recognizes in his calculations it has long been recognized that a ductile material is desirable in order to equalize abnormal stresses caused by eccentric loading, notches, impact loading, and overstrain. Weld metal must possess the property of ductility in order that these abnormal stress conditions may be met.

Characteristics of Electrode Coverings.

The coverings which are applied to welding electrodes, in order to produce the desirable properties detailed in the previous section, vary greatly in their chemistry and in the methods used for their application to the core wire. However, there has been a noticeable standardization, both in arrangement of these covering compositions into well defined types and in regard to the mechanical handling of the coverings in processing.

While dipping and winding of covering materials is still practiced to some extent, by far the majority of welding electrodes are covered by the process of extrusion. Briefly, this process involves forcing an annulus of plastic covering material around the core wire by means of a hydraulic press, gear pump or the like. After the covering has been placed on the core wire, excess moisture is driven off by a baking process.

It would be extremely desirable to have a single type of shielded arc welding electrode which would perform satisfactorily all the

types of welding which are encountered in industrial and field applications. If this were possible, such an electrode would be required to perform satisfactorily with A.C. as well as D.C. welding current; it would be suitable for welding in the vertical and overhead positions, as well as in the flat position; it would produce dense welds as measured by X-ray examination, and it also would produce an excellent surface finish. These four features are not all obtainable in their highest state of development in one electrode, however. The coverings which tend to produce excellent A.C. operation do not usually work satisfactorily in the vertical and overhead positions; also the features of X-ray density and surface finish are usually attained at some sacrifice of operation in the vertical and overhead positions. Since all of these factors are important, there has been a tendency to develop three distinct classes of electrodes, each of which favors one of the main requirements at a partial sacrifice of the other requirements.

In describing the three classes of electrode coverings, the nomenclature used in the Navy Department specification for welding electrodes will be followed. This specification classifies all-position electrodes as class 1, electrodes suitable for making welds in the flat position as well as horizontal fillets as class 2 electrodes, and electrodes suitable for welding in the flat position only, as class 3 electrodes.

CLASS 1 ELECTRODES.

Electrodes of this class are ordinarily of the cellulosic type. They are suitable for welding in the flat, vertical, horizontal, and overhead positions and operate best when attached to the positive pole of the welding generator. The principal reason for the almost universal use of cellulosic coverings for this class of work is that vertical and overhead operation is most successful when a limited amount of slag is deposited by the electrode. In order to obtain satisfactory shielding of the weld metal with this small slag volume the use of a gas shield is indicated. Class 1 electrodes are therefore characterized by relatively thin covering, usually less than 12 per cent of the weight of the core wire. Although numerous combinations of ingredients are used to develop the necessary character-

istics for all-position welding, the following chemicals are frequently used in such mixtures: finely divided cellulose, titanium oxide (rutile), ferro-manganese, and sodium silicate. The function of the cellulose is to produce a gas shield which excludes the atmosphere from the arc space. Titanium oxide has two particularly desirable properties which make it the most universal metallic oxide used in electrode coatings. Its ionization characteristics are such that it reduces the arc voltage of electrodes in which it is used. For vertical and overhead deposition a low arc voltage is desirable to avoid overheating of the metal. Titanium oxide slags have the necessary combination of density and fluidity under arc conditions to avoid slag entrapment when welding in the vertical and overhead positions. Ferro-manganese is added as a deoxidizer and also for the purpose of reducing the number of blow holes in the weld. In the absence of ferro-manganese a considerable portion of the manganese of the core wire is lost in passing through the arc. The added ferro-manganese acts to restore in the weld metal the normal manganese required in high quality steel. Sodium silicate is universally used as one of the important ingredients of welding electrodes. Sodium silicate has a variety of functions. First it acts as a binder for the finely divided dry materials of the coating. For the operation of extrusion, a plastic mass of the consistency of a heavy dough is required. Sodium silicate confers the necessary plasticity to the mixture so that a uniform covering may be extruded upon the electrode. During welding the sodium silicate acts as a combustion retarder for the combustible materials of the coating, with the result that the covering projects a short distance beyond the tip of the burning core wire. This projecting portion of the covering is like an inverted crucible, and acts to direct the arc. Finally sodium silicate combines with the other metallic oxides to form a welding slag similar in composition to acid open hearth steel slags.

CLASS 2 ELECTRODES.

In formulating the coverings for this class of electrodes it is necessary to produce electrodes which in addition to the basic requirements of making a horizontal fillet weld, will operate equally satisfactorily with A.C. and D.C. welding current, and will also

permit a higher rate of deposit than is possible with the class 1 electrodes. To meet these requirements the amount of cellulosic material is usually decreased and a larger proportion of metallic oxides is used. Some of the basic ingredients, whose functions were described in the preceding paragraphs, are usually present also in class 2 electrodes, but in revised percentages. The quantity of coating used on these electrodes is usually between 10 and 15 per cent of the weight of the core wire. Since these electrodes are suitable forA.C. welding, they usually operate equally well when attached to the positive or negative pole of the welding generator, in D.C. welding.

CLASS 3 ELECTRODES.

Inasmuch as these electrodes are intended for use in the flat position only, much more fluid slag and greater volumes of slag may be used. The covering of electrodes of this type contains little or no combustible material. These coverings are usually very heavy, of the order of 20 per cent of the weight of the core wire. Sodium silicate and ferro-manganese are almost always present in these coverings as well as in the class 1 and class 2 electrode coverings. Titanium oxide may be present in some cases. The remainder of the composition is ordinarily made up of oxides of iron, manganese and aluminum. Because of the absence of cellulosic material in electrodes of this class these electrodes may be operated at very high current densities resulting in increased welding speeds. In addition to this feature of greater economy, the class 3 electrodes produce welds of greater density as measured by X-ray examination.

TABLE - 4

PHYSICAL PROPERTIES OF SHIELDED ARC WELD METALS.

	Class 1 All Position (as welded)	Class 2 Horizontal Fillets (as welded)	Class 3 Flat Positions Only	
			As welded	Stress Relieved
Yield Point	55,000 psi	55,000 psi	55,000	50,000
Ult. Strength	65,000 psi	65,000 psi	65,000	60,000
% Elongation - 2"	18%	22%	25%	35%

Table 4 gives the comparative physical properties of the deposited weld metal of the three classes. It will be noted that the ductility is greatest for class 3 weld metal. In order to satisfy the specialized requirements for position welding, vertical and overhead, certain concessions must be made regarding the metal quality. The greater coating thickness on the class 3 electrodes is a factor in obtaining the most desirable physical properties. For position welding smaller coating thicknesses are mandatory to satisfy the special workability requirements in the vertical and overhead positions.

The All-Weld Metal Tension Test.

In reporting the physical properties of metal deposited by various electrodes the values which have been given in the tables are obtained from tests made on samples consisting entirely of weld metal. For evaluating the characteristics of various electrodes this test is of primary importance, since it is least affected by the properties of the metal being welded. While other tests of weld metal are frequently conducted the tensile strength, ductility, and the yield point are of primary interest.

In testing electrodes it is customary to determine the properties of the weld metal in the as-welded condition. The properties of the as-welded weld metal are of interest even though the structures on which the electrodes are to be used will be stress relieved after welding. In such structures it is important to have electrodes which are ductile in the as-deposited state so that they may resist the cooling stresses which accompany all welding operations. A large number of structures are of such a character that it is impracticable to stress-relieve them. In such structures the as-welded condition is the condition in which the weld metal will be used. Excepting for a slight decrease in tensile strength and yield point the operation of stress-relieving has little effect on the evaluation of electrodes. Inasmuch as the test results in the as-welded condition are the critical ones, these tests are the ones which are ordinarily made.

As stated before, the all-weld metal tension test is the least affected by the properties of the base metal of any welding test.

However, it is of extreme importance to standardize the conditions of test in order that reproducible results may be obtained. The principal variables are the technique of deposition (beading and weaving) and the temperature of the base metal.

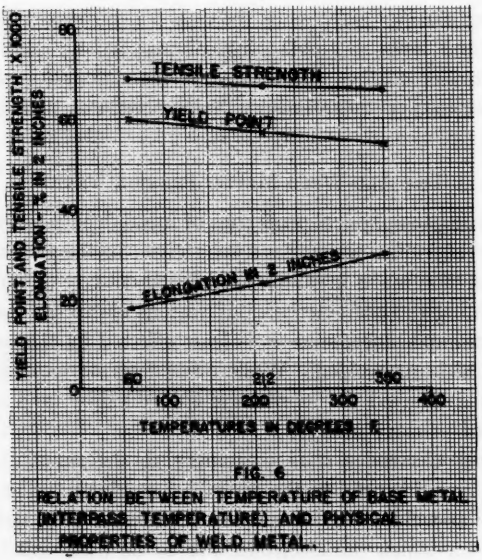


FIG. 6

RELATION BETWEEN TEMPERATURE OF BASE METAL (INTERPASS TEMPERATURE) AND PHYSICAL PROPERTIES OF WELD METAL.

Figure 6 illustrates the relation between the temperature of the base metal and the physical properties of deposited weld metal. It will be noted that the ductility of weld metal deposited on preheated base metal is considerably higher than the ductility of weld-metal deposited on cold base metal. Figure 7 illustrates the relation between beading and weaving technique on the properties of deposited weld metal. Metal deposited in straight beads is ordinarily not as ductile as metal deposited with the same electrodes using a weaving technique.

In addition to the variables discussed the question of the time of aging has frequently been suggested as an important variable. It was believed that the elapsed time between welding and machining

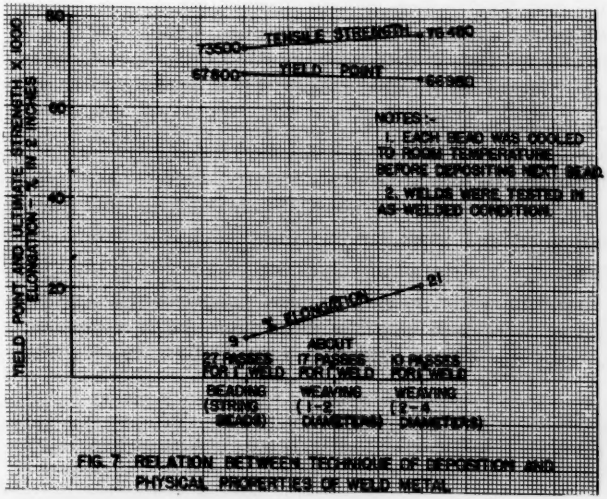

TENSILE STRENGTH
YIELD POINT
NOTES :-
1. EACH BEAD WAS COOLED TO ROOM TEMPERATURE BEFORE DEPOSITING NEXT BEAD.
2. WELDS WERE TESTED IN AS-WELDED CONDITION.
% ELONGATION
ABOUT
27 PASSES FOR 1" WELD
BEADING (STRING BEAD)
17 PASSES FOR 1" WELD
WEAVING (1-2 DIAMETERS)
10 PASSES FOR 1" WELD
WEAVING (2-4 DIAMETERS)

FIG. 7 RELATION BETWEEN TECHNIQUE OF DEPOSITION AND PHYSICAL PROPERTIES OF WELD METAL

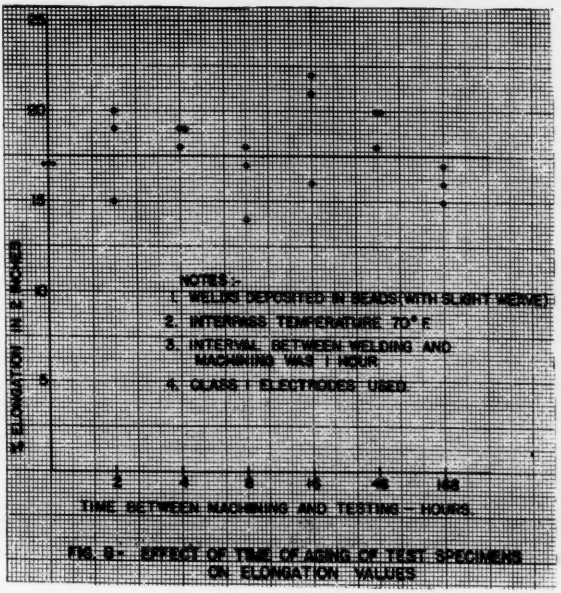

NOTES :-
1. WELDS DEPOSITED IN BEADS (WITH SLIGHT WEAVE).
2. INTERPASS TEMPERATURE 70° F.
3. INTERVAL BETWEEN WELDING AND MACHINING WAS 1 HOUR.
4. CLASS I ELECTRODES USED.
TIME BETWEEN MACHINING AND TESTING — HOURS.

FIG. 8.- EFFECT OF TIME OF AGING OF TEST SPECIMENS ON ELONGATION VALUES

or the total elapsed time between welding and testing might have an important influence on the test results. Figure 8 shows the results of some tests to determine the effect of time of aging on the physical properties of weld metal. There appears to be no significant change in properties with time. However, the test results collected in this investigation when analyzed statistically bring out an important feature of electrode testing. Figure 9 represents the distribution of values of ductility obtained in this particular investigation and a similar one using slightly different welding conditions. This spread of values was obtained even though all of the important variables such as plate temperature and deposition technique were kept constant throughout the investigation. In order that the reproducibility of results may be kept within the

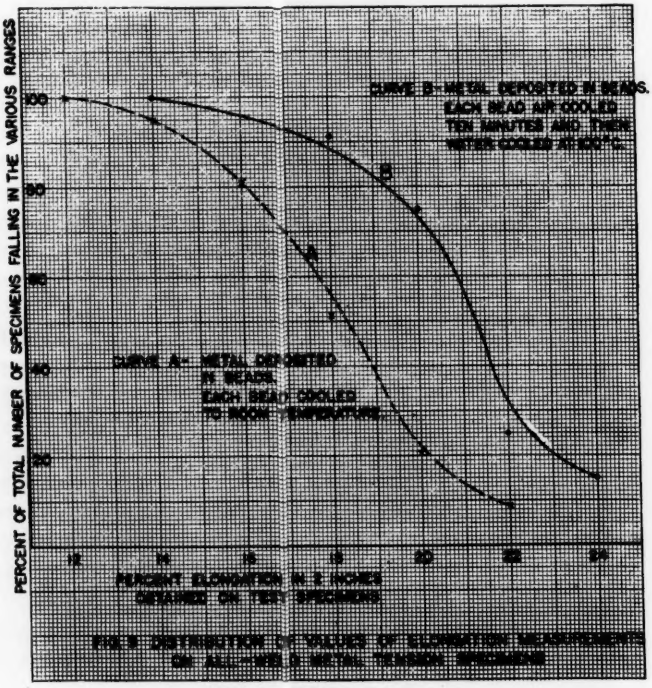


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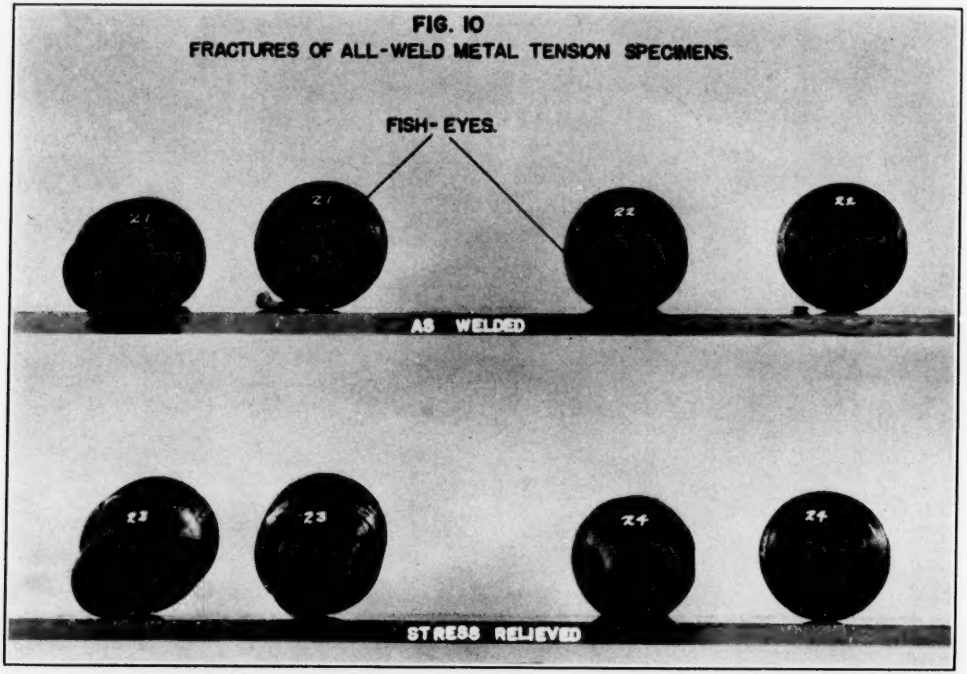
FIG. 10

FRACTURES OF ALL-WELD METAL TENSION SPECIMENS.



limits found in this investigation it is extremely important that all variables be kept constant.

The large variation in elongation values is not caused by improper manipulation as might be suspected, but rather by the peculiar type of fracture exhibited by as-welded deposits of class 1 electrodes. Circular patches in the fracture, commonly referred to as "fish-eyes" tend to reduce the normal ductility of the metal. The amount of the reduction is dependent on the size and number of the "fish-eyes." These patches are not necessarily the result of defective weld metal. This statement is substantiated by the fact that welds made with similar electrodes under identical conditions of deposition are free of all defects when tested after stress-relieving. See Figure 10. It is believed that the "fish-eyes" result from a local tension fracture occurring at a non-metallic inclusion or blowhole; which tension fracture proceeds radially from the defect for a short distance before the more normal shear type of fracture occurs. The phenomenon is very likely also associated with the presence of locked-up stresses in the as-welded deposits.

Other Tests of Electrodes.

While the properties of deposited metal are of primary concern in the testing of covered electrodes there are certain characteristics of the coverings themselves which must be carefully controlled if satisfactory results are to be obtained. It is important that the covering be applied to the core wire in such a way that it is concentric with the core wire. Certain methods of covering electrodes have been developed specifically to overcome the tendency for eccentricity. In the process of extrusion which is most frequently used in the covering of electrodes eccentricity of the coverings is an ever present possibility, and the coverings must be rigidly checked to be certain that the eccentricity does not exceed a specified amount. Electrodes with eccentric coverings burn irregularly, frequently causing the heavy side of the covering to extend far beyond the burning core wire. This defect in operation is commonly referred to as "tailing."

In humid climates such as are encountered in ship building yards, electrode coverings are likely to absorb considerable moisture from

the atmosphere during wet weather. Although it is desirable that electrodes operate equally well in wet or dry weather, it has been found necessary to accept the fact that the cellulosic type of electrode will not operate as well when it is very wet as it does when it is dry. Storage ovens are usually provided to maintain large stocks of electrodes at suitable temperature and humidity. With electrodes of this type it is usually required that the coverings will not be damaged by alternate humidification and drying. Satisfactory welds must be made after the rods have been humidified and dried several times.

Another important test of electrodes is the fillet weld test. This test is often referred to as the usability test, and is made to determine the ability of the electrode to deposit fillet welds of good contour, free from defects such as cracks, surface porosity, and undercutting.

Bend tests are frequently made transverse to welds to determine whether the electrodes are capable of producing a satisfactory bond between weld-metal and base-metal. Face bend and root bend tests are employed for this purpose. In these tests the name signifies that surface of the plate which is in tension.

Inasmuch as certain classes of welded work are required to be radiographed it is important that welding electrodes used for such work be capable of depositing weld metal of the required radiographic soundness. Tests of electrodes for general use are therefore required to pass radiographic requirements in order that they may be suitable for all classes of work.

Tests which are less frequently used in the testing of electrodes include the transverse tension test, the notch bar impact test and the tension impact test.

Certain other tests used to evaluate the economy of welding include tests for slag removal, rate of deposition tests, and depositing efficiency tests. In the slag removal tests, the time required to clean off the slag from a specified length of welded bead is determined. It is required that the slag be removable by means of hand tools only. The efficiency and rate of deposit tests are made simultaneously by carefully weighing the electrode consumed and the weld metal deposited in a measured length of time. The efficiency

of deposit is an indication of the spatter loss of the electrode. The rate of deposition is a measure of the economy of using the electrode. The feature of high deposition rate is the principal advantage in the use of class 3 electrodes over the class 1 and class 2 electrodes.

Low Alloy Steel Electrodes.

Covered metallic arc electrodes are available for the welding of a variety of compositions of low alloy steels which have recently been introduced for structural, high temperature, and other applications. The coverings for low alloy steel electrodes are quite similar in composition to those used on carbon steel electrodes. There are several methods of producing low alloy steel weld metal. The simplest method is to use a core wire of the composition which it is desired to deposit. In this case the covering may be identical with the covering used on carbon steel electrodes. Frequently, however, the alloys are added to the covering, either in the form of ferro-alloys or metallic oxides. In the cellulosic type of coverings it is possible to reduce the oxides of metals such as nickel, copper and molybdenum to the metallic form without much loss of these metals to the slag. This method of adding alloys to the covering is definitely limited because large percentages of alloy in the covering result in covering thicknesses which make the electrode difficult to handle.

Alloy steel weld metal has a higher strength than carbon steel, and consequently, it tends to have a somewhat lower ductility. Accordingly, it is desirable in testing low alloy steel electrodes to supplement the tests already described in the previous section with a special test to evaluate the ductility of the deposited metal throughout the temperature range of cooling. This may be most conveniently done by measuring the susceptibility of the alloy weld metal to cracking under restrained conditions. Figure 11 illustrates a simple type of cracking test proposed by Swinden and Reeve for testing of alloy steel electrodes. In this test two plates which are to be welded together are rigidly bolted to a strongback. The upper plate is welded on three sides to the lower plate by means of fillet welds. The assembly is then permitted to cool

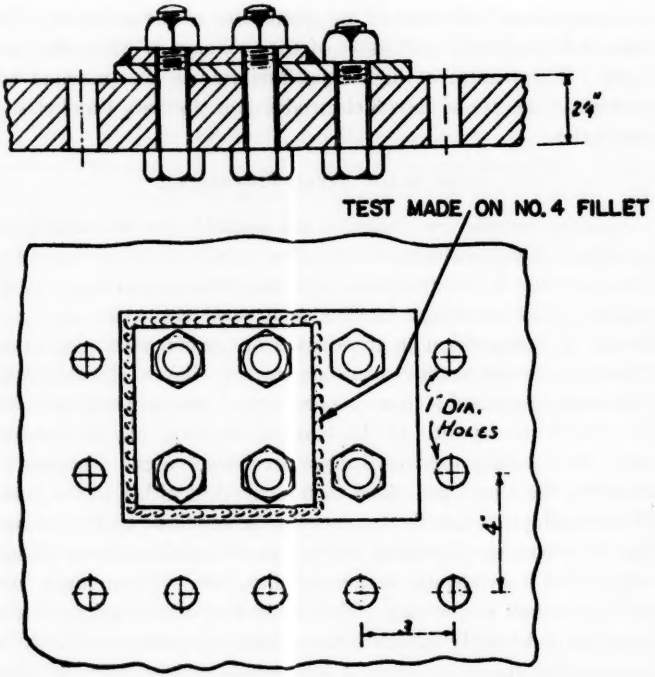


FIG. 11 WELD CRACKING TEST (SWINDEN & REEVE)

to room temperature. After cooling, the fourth fillet weld is made using the electrodes which are to be tested. This fourth fillet weld is then examined for cracks.

For certain alloy steel electrodes welding under fixed conditions and at room temperature may be too severe a test. If cracks are found in fillet welds made under these conditions, it may still be possible to make satisfactory welds using these electrodes. Preheating of the base metal will reduce the hardness and the susceptibility to cracking of alloy welds. Since the root bead, that is, the first bead which is deposited, is the most likely to crack, it is good practice to use a carbon steel electrode of the maximum ductility for depositing the root bead. Subsequent beads may then be deposited with the alloy steel electrodes. In order to avoid

cracking in alloy steel welds it may also be desirable to decrease the speed of welding. Anything which tends to decrease the rate of cooling will overcome the tendency toward hardening and the formation of cracks.

TABLE 5 - PHYSICAL PROPERTIES OF WELD METAL DEPOSITED WITH VARIOUS LOW ALLOY STEEL ELECTRODES.

Kind	Physical Properties					
	As welded			Stress relieved		
	Y.P.	Ult.	% El. 2"	Y.P.	Ult.	% El. 2"
Carbon-molybdenum .50	65,000	75,000	20%	54,000	70,000	30%
Ni-Cu steel (Foley)	60,000	80,000	20%	65,000	80,000	25%
Cu-Ni-Mo steel (RDB)	60,000	80,000	20%	65,000	80,000	25%
Cu-Cr-P steel (Cortes)	70,000	80,000	20%	65,000	78,000	22%
Chrome-copper steel	70,000	80,000	20%	65,000	78,000	22%
2% Nickel steel	70,000	85,000	20%	65,000	80,000	25%
Croomall steel	85,000	95,000	16%	80,000	90,000	23%
5% Chrome-molybdenum steel	75,000	85,000	*20%	70,000	80,000	26%

*Preheating required to obtain these properties.

Table 5 illustrates typical compositions of low alloy steel electrodes and the physical properties obtainable with these electrodes in the as-welded and the stress relieved conditions.

Stainless and Heat Resistant Alloys.

Covered electrodes are also available for welding the majority of the commercial types of corrosion resisting and heat resisting alloy steels. These steels are either ferritic alloys of iron and chromium or austenitic alloys of iron, chromium and nickel. The most important of this group of alloys is the austenitic alloy commonly known as "18-8" containing 18 per cent chromium and 8 per cent nickel. Since the corrosion resistance of the weldable grades of these alloys depends on the maintenance of a low carbon content, electrode coverings for these alloy steels are usually free from cellulosic or other carbonaceous materials, or employ such substances in small amounts. In addition to the usual oxides of silicon and aluminum, these coverings usually contain appreciable percentages of fluorides.

Tests of stainless steel welding electrodes usually include corrosion tests of the deposited weld metal. These corrosion tests are usually comparative, having as their object the demonstration that the deposited weld metal is as resistant to a certain corrosion environment as the base metal. Welds in the straight chromium irons must be made with great care, in order to avoid cracking. These alloys are air hardening up to 15 per cent chromium. Above 15 per cent, the deposited weld metal is not very hard but is extremely coarse grained, and must therefore be handled carefully to avoid cracks.

Frequently in the welding of 18-8 alloy steel it is found necessary to join the alloy material to carbon steel structures. When this is necessary, it is preferable to use an electrode containing somewhat higher chromium and nickel content than the basic 18-8 composition. This is desirable because of the dilution of the alloy weld metal with carbon steel from the base metal. Either 25 chromium—12 nickel or 25 chromium—20 nickel electrodes may be used for this purpose. Either of these electrodes will deposit a completely austenitic, and therefore, a ductile weld.

Table 6 is a tabulation of the types of corrosion resisting steel welding electrodes which are available commercially. The properties of the 13 per cent chromium steel weld metal are dependent on preheat and postheat treatments. The weld metal may be hard-

TABLE 6- KINDS OF COVERED CORROSION RESISTING STEEL ELECTRODES COMMERCIALLY AVAILABLE.

Chemical Abbreviation	Iron and Steel Institute Number	Chemical Analysis			
		Chromium	Nickel	Carbon	Other Elements
		%	%	%	%
13 Cr	410	12-14		.12 max.	
18 Cr	430	16-18		.12 "	
28 Cr	446	23-30		.35 "	
*18-8S	304	17-19	7-9	.08 "	
18-8S-Ti		17-19	7-9	.08 "	Ti 4XC
18-8S-Cb		17-19	7-9	.08 "	Cb 10XC
18-8S-Mo	316	17-19	7-9	.11 "	Mo 2-4
25-12	309	22-26	11-13	.20 "	
25-20	310	24-26	19-21	.25 "	
20-25	311	19-21	24-26	.25 "	
29-9	312	27-31	8-10	.25 "	
15-35	330	14-16	33-36	.25 "	
15-60		14-16	58-62		
15-80		14-16	78-82		

*In the iron-chromium-nickel alloys, the first number indicates the chromium content and the second number the nickel content. The "S" indicates low carbon.

ened and softened by heat-treatment in a manner similar to 13 per cent chromium steel base metal. Under carefully controlled procedure, the 18 per cent chromium and 28 per cent chromium steel electrodes will also deposit ductile weld metal, of strength equivalent to that of the respective base metals.

All of the austenitic types (chromium-nickel) produce weld metal having a tensile strength over 75000 pounds per square inch, a yield point over 35000 pounds per square inch, and an elongation in two inches over 25 per cent.

Hard Facing Alloys.

In designing a machine part it is often desirable to produce a hard wearing surface on a tougher and softer center or core. Met-

allurgical methods such as case carburizing and chromium plating, have been used extensively to accomplish this purpose. By the proper selection of alloys, a series of hard facing deposits may be applied to metals by welding, varying in hardness from a few points Brinell above the base metal to the hardness of Carboloy. For certain applications where a machinable surface somewhat harder than the base metal is desired, low alloy steel electrodes containing chromium, nickel, copper, molybdenum, and vanadium as alloy elements are employed. These electrodes have been described above under Low Alloy Steel Electrodes.

Electrodes of the chromium nickel austenitic types are frequently used for hard facing applications because they work-harden rapidly in service and because they possess corrosion resistance in addition to hardness, making them quite useful for applications where erosion is encountered. A good example of this use of austenitic chromium-nickel electrodes is the hard facing of steel water turbine rotors. Austenitic manganese steels are also available in electrode form and are widely used in depositing wearing surfaces on earth-handling machinery and battered rail ends.

There are also available a variety of proprietary electrodes for depositing controlled hardnesses of alloy metal to take care of the need for non-machinable hard facing alloys ranging from about 35 Rockwell-C hardness to 70 Rockwell-C. These metals obtain their hardness from carbides or borides, of elements such as manganese, tungsten, chromium, and molybdenum. Electrodes depositing high speed steel of the 18 tungsten, 4 chromium, 1 vanadium, analysis represent one of the simpler analyses of this group.

Some of the most useful hard facing alloys give the best results when deposited by means of oxy-acetylene welding. The chromium-cobalt-tungsten alloy used for hard facing of seats and disks of steam valves is a good example of this type of alloy. While it is feasible to deposit this alloy by means of metal arc welding, the oxy-acetylene process is to be preferred. With the latter method, the metal can be sweated onto the base metal with a minimum of contamination with iron. Since iron tends to decrease the hardness and wear resistance of the chromium-cobalt alloy, the dilution of the alloy with iron from the base metal should be avoided.

TABLE 7 - COMPOSITIONS AND HARDNESSES OF SOME HARD-FACING METALS DEPOSITED BY ARC WELDING.

Kind	Chemical Analysis					Brinell Hardness	
	Cr	Ni	V	Mn	Other	As Deposited	Work Hardened
Ni-Cu Steel		2.00			Cu 1.00	160	
Cr-Mn-Si Steel	.80			1.00	Si .75	190	
18 Cr - 8 Ni	18.0	8.0				175	450
16 Cr - 6 Ni	16.0	6.0				200	525
18 Cr	18.0					375	425
Hi-speed steel	4		1		W 18.0	625	
Manganese steel		5.0		12.0		200	500
Chromium boride		75.0	+ Chromium borides			550	
Self-hardening	Granular carbides in low carbon steel sheath					500	

Table 7 lists typical electrodes of the hard facing types with their approximate compositions and the hardness which may be expected in the deposited metal. In all these deposits it is assumed that the base metal is a low carbon steel. While the hardnesses are very useful in classifying the various types of metal, it should not be assumed that the harder the metal the better the wear properties. Wear is a complex phenomenon which depends on factors other than hardness. Accordingly, the selection of a particular type of hard facing metal should be based upon its performance under similar or identical service conditions.

Electrodes for Welding of Cast Iron.

The welding of cast iron presents certain difficulties because of the inherent brittleness of the base metal. Most of the methods of welding cast iron offer a sort of compromise between the desire to deposit metal of similar composition to that of the base metal and the necessity to deposit weld metal of sufficient ductility to absorb the welding stresses. One type of electrode which is fre-

quently used consists of a very low carbon steel core wire coated with materials which tend to decrease the penetration of the weld metal into the cast iron. By limiting the penetration, the carbon content of the deposited metal is maintained at a sufficiently low value to remain ductile. Welds made with such electrodes are frequently reinforced by means of studding.

Covered electrodes depositing monel, nickel, or austenitic chromium nickel steel may also be used for welding cast iron. With these alloys the penetration into the base metal does not appreciably affect the ductility of the deposited metal. Accordingly, machinable welds may be deposited with these types of electrodes.

Covered cast iron electrodes are also available for welding of cast iron. These electrodes require the local pre-heating of the base metal to approximately 700 to 800 degrees F. in order that a sound deposit may be obtained. These electrodes produce a weld similar in chemical analysis to the base metal, and may therefore be preferred in certain applications over the more ductile weld metals mentioned in the preceding paragraphs.

Non-Ferrous Alloys.

Most welding of copper base alloys is done by the oxy-acetylene and carbon arc welding processes. However, in recent years a limited number of coated copper-base alloy electrodes have been developed. With the single exception of the 70-30 copper nickel alloy, these coated electrodes employ only a thin coating of protective materials which is applied by dipping. The 70-30 copper nickel electrode has a thicker covering applied by extrusion in a manner similar to the application of coverings to steel electrodes.

Covered electrodes have been developed for the welding of nickel copper alloy (monel metal), and nickel. Weld metal of very satisfactory strength and ductility which closely matches the chemical analysis of the base metal may be deposited with these electrodes. The electrode coverings on these compositions are also quite heavy and are applied to the wire by extrusion. Corrosion tests of welded specimens in various media, including sea water, indicate that the corrosion resistance of the weld is equivalent to that of the base metal for the nickel-base alloys as well as the 70-30 copper nickel alloy.

The metal arc welding of aluminum and its alloys is performed by the use of a coated 5 per cent silicon-aluminum alloy electrode. The coatings usually consist of mixtures of chlorides and fluorides, and are applied by dipping.

The compositions and properties of the weld metals deposited with the various copper-, nickel-, and aluminum-base alloy electrodes are given in Table 8.

TABLE 8- COMPOSITIONS AND PHYSICAL PROPERTIES OF NON-FERROUS WELD METALS.

Kind	Chemical Analysis						Physical Properties (as welded)		
	Cu	Ni	Al	Sn	Si	Other	Y.P.	Ult.	% Elong - 2"
	Copper - base alloys						psi	psi	
Phosphor bronze	93.5			6		.10	20,000	35,000	10.0
Silicon bronze	96				3	Mn 1	20,000	45,000	15.0
70:30 Copper nickel	70	30					30,000	42,500	30.
	Nickel - base alloys								
Nickel copper alloy	30	70					35,000	70,000	25.
Nickel		99+					30,000	60,000	25.
	Aluminum - base alloy								
5% Silicon aluminum			95		5			15,000	2%

SUMMARY.

This discussion has endeavored to show how the use of coverings on welding electrodes has improved the quality of welds made with the metal arc process. The principal improvements achieved by the use of coverings are the following:

1—Improved ductility and tensile strength of deposited metal.
2—Greater arc stability.
3—Faster speeds of welding.
4—Better penetration into corners.
5—Improved surface appearance.

By the use of coverings a variety of arc characteristics may be produced. No one electrode covering can be expected to produce the maximum ductility and X-ray soundness and at the same time be suitable for vertical and overhead welding and for use with alternating current. Present developments have produced one class of electrodes which is suitable for all position welding using direct current. A second class may be used with alternating as well as direct current for the welding of fillet and butt welds in the flat position and fillets in the horizontal position. A third class, also suitable for A.C. as well as D.C. welding may be used for flat position welding only. This class of electrodes usually has the highest ductility and X-ray soundness obtainable with covered electrodes.

Tests of the properties of weld metal, particularly when made in the as-welded condition, may be expected to yield reproducible results only when the welding conditions are rigidly controlled. Test results have been presented to show the importance of controlling factors such as interpass temperature and welding technique (beading or weaving).

Covered electrodes are available in a large variety of core wire compositions. The developments of the last decade have made available electrodes for metal arc welding almost every important metal used for construction purposes.

Acknowledgments.

In conclusion the author wishes to acknowledge the assistance of Mr. J. L. Cahill who conducted the tests concerning the influence of the several variables on the properties of deposited weld metal, and Mr. Bela Ronay who conducted the tests on the carbon-molybdenum steel and some of the non-ferrous alloy electrodes.

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A DERIVATION OF FROUDE'S QUASI-DIMENSIONLESS COEFFICIENT $B_p = \frac{NP^{1/2}}{V^{5/2}}$ AND SOME COMMENTS ON ITS USE AS A BASIS FOR PROPELLER DESIGN.

By Lieutenant Commander J. M. Labberton, U. S. N. R.*

Present day propeller designers make use of two quasi-dimensionless coefficients, B_p and B_u, which seem to be shrouded in mystery, since apparently their derivation has never been published. To dissipate some of that mystery a brief history of those coefficients and an independent derivation of them will be presented in this paper.

Admiral Taylor, in his book, "The Speed and Power of Ships" (1)† says that the coefficient $B = \frac{\text{Revolutions} \times (\text{Power})^{1/2}}{(\text{Speed})^{5/2}}$ was first proposed by Froude in 1890. This proposal, which was more in the nature of a statement as to what the British Admiralty Experiment Works was doing, was made by Mr. R. Edmund Froude (the younger) in discussing a paper by Mr. S. W. Barnaby on "The Screw Propeller" (2) in 1890. In part, he said:

"In reference to the form in which the author (Barnaby) had presented those results (of propeller tests) for use, his feeling was that he was indebted to anyone who would do as the author had done, and not only make use of the results, but also find new and convenient forms into which they might be thrown for purposes of use. And those persons were the most competent to give an opinion upon the form most convenient for use, who were professionally employed in determining the dimensions of propellers suitable for various conditions."

* Asst. Professor of Mechanical Engineering, New York University.
† Numbers in parentheses refer to bibliography at the end of this paper.

He then went on to say that the Admiralty Experiment Works had first used a form which was "afterwards practically discarded in favor of another which bore more resemblance to the author's form. It consisted of two constants, O_R and O_D, as follows

$$O_R = \frac{\text{Tens of Revolutions per Minute} \times (\text{IHP})^{1/2}}{(\text{Speed in Tens of Knots})^{5/2}}$$

$$O_D = \frac{\text{Diameter} \times (\text{Speed in Tens of Knots})^{3/2}}{(\text{IHP})^{1/2}}$$

Froude makes no mention of the derivation of these "constants," if indeed, a mathematical derivation existed. It can be seen at a glance that O_R is, in effect, our B_p or B_u of today.

Later in his discussion Froude says:

"Of course, the form into which results of that kind should be thrown for practical purposes depended very much upon the shape which the problem took. The shape in which it generally presented itself in the Admiralty Experiment Works was this:

"A new ship having been designed, the intended speed being fixed upon, and the HP corresponding to that speed having been determined by experiment, the revolutions were next decided by the engineers, so that the proposed HP might be most advantageously developed by the type of engine intended to be employed. Consequently, there were the speed, the HP, and the revolutions, as fixed quantities from which to work."

As we who have been confronted with propeller design well know, the problem is little different today—especially that feature "the revolutions decided by the engineers"—meaning a group not responsible for propeller performance, but only for engine (or turbine) performance.

In the case of any screw propeller working in a viscous fluid, Taylor (3) presents a formula for propeller thrust which he indicates was derived by dimensional analysis, and Buckingham (4) has recently presented the full derivation of the formula which is as follows:

$$F = d^2 \rho V^2 \left[f\left(\frac{nd}{V}\right), \frac{\nu}{dV}, \frac{dg}{V^2} \right]$$

Where

F = Propeller thrust in pounds.

d = Propeller diameter in feet.

n = Propeller revolutions per second.

V = Propeller speed of advance in feet per second.

ρ = Density of fluid in pounds per cubic foot.

ν = Kinematic viscosity of fluid.

g = Gravity acceleration.

Taylor indicated and Buckingham discussed in detail the reasons why we are justified in ignoring the terms corresponding to Reynold's number (inverted) and Froude's number *when the propeller is not in cavitation.* Doing this, we obtain the general expression for any propeller *not in cavitation* as

$$F = \rho\, d^2\, V^2 \times f\left(\frac{nd}{V}\right)$$

It is obvious that $\frac{nd}{V}$ is a dimensionless quantity denoting slip and is independent of the size of the propeller.

We can write the equation for power as: Power = P = zFV, where "z" is a dimensionless constant.

Let p = propeller pitch in feet. Then for a given pitch ratio, p/d, the slip function, f(nd/V) is a constant for any given speed, V. Let (nd/V) = δ. Then substituting the value for F in the equation for power we get:

$$P = z\, \rho\, d^2\, V^3 \times f\,(\delta)$$

But $\delta = \frac{nd}{V}$ or $d = \delta\, \frac{V}{n}$

$$\therefore P = z\, \delta^2\, f\,(\delta)\, \rho\, \frac{V^2}{n^2}\, V^3 = z\, \delta^2\, f\,(\delta)\, \rho\, \frac{V^5}{n^2}$$

Let $z\, \delta^2\, f\,(\delta) = B^2$ which is dimensionless

$$P = B^2 \rho \frac{V^5}{n^2}$$

$$B^2 = \frac{n^2 P}{\rho V^5}$$

$$B = \frac{n P^{1/2}}{\rho^{1/2} V^{5/2}}$$

which is a dimensionless coefficient fixing the performance of any propeller regardless of size but having a given pitch ratio, blade proportions, etc., and *at a given slip but not in cavitation.*

However, since water is the only fluid with which we are concerned at present, we can dispense with the variable ρ, and use a quasi-dimensionless coefficient. Moreover, since it is customary in propeller problems to use "knots" instead of "feet per second"; and "revolutions per minute" instead of "revolutions per second", we will hereafter let V = knots, and N = RPM. Our quasi-dimensionless coefficient then becomes:

$$B_p = \frac{N P^{1/2}}{V^{5/2}}$$

when the subscript "p" indicates the coefficient is concerned with the power input of the propeller rather than the useful power delivered.

Therefore, if we test any model propeller at a given slip and measure

Revolutions per Minute
Power Input in Horsepower
Speed of Advance in Knots

we can determine the coefficient B_p which will hold good for any size of propeller of geometrically similar proportions.

Since we now have a practical coefficient which we can use incorporating nautical terms, such as, RPM, Horsepower and Knots, let us work out an expression for slip involving $\delta = \frac{nd}{V}$ and a = pitch ratio or $a = \frac{p}{d}$, where "p" = pitch of propeller in feet.

First we will define S = slip ratio, as that percentage of a propeller revolution turned which does not advance the propeller forward in proportion to pitch, therefore, the propeller should advance in feet per minute.

$$n\ p\ (1 - S) = nad\ (1 - S)$$

If V = Knots, then 101.33 V = feet per minute.

$$101.33\ V = nad\ (1 - S)$$

$$V = \frac{nad\ (1 - S)}{101.33}$$

$$\delta = \frac{nd}{V}$$

$$\delta = \frac{nd\ (101.33)}{nad\ (1 - S)}$$

$$\delta = \frac{101.33}{a\ (1 - S)}$$

$$1 - S = \frac{101.33}{a\ \delta}$$

Thus, knowing pitch ratio and δ, slip S may be determined.

Consider a series of propellers, all having the same number of blades, the same thickness of blade in proportion to diameter, the same width and contour of blade in proportion to diameter, in short, all of these propellers the same, in proportion, except pitch ratio. It is then obvious that if these be run at a given power input, a given revolutions, and a given speed of advance, that the value $\delta = \frac{nd}{V}$ will be different for each pitch ratio, decreasing as pitch ratio increases and vice-versa. If the efficiency of the propellers is determined at the same time, it is obvious that these data can then be used to determine or predict the performance of any propeller of any size or speed provided the coefficient $B_p = \frac{NP^{1/2}}{V^{5/2}}$

is of the same value as tested and provided the propeller is not in cavitation. Similarly, if various other values of B_p are tested over a sufficiently wide range, we can then predict the performance of this type of propeller in any service and of any pitch ratio and size, retaining one proviso of staying out of cavitation.

Rear Admiral D. W. Taylor (C. C.), U. S. N., has plotted such a series of tests and it is shown as Figure 204 in his book, "The Speed and Power of Ships."

From the above it follows, that in a similar manner, the power output, U, of the propeller could be used as well as the power input, P, and a similar coefficient $B_u = \frac{N\ U^{1/2}}{V^{5/2}}$ be used, the only difference being the factor of the efficiency of the propeller.

If e_p = propeller efficiency then

$$e_p\ P = U$$

$$\text{or } P = \frac{U}{e_p}$$

$$B_p = \frac{N\ P^{1/2}}{V^{5/2}} = \frac{N\ U^{1/2}}{e_p^{1/2}\ V^{5/2}}$$

$$B_p\ (\sqrt{e_p}) = \frac{N\ U^{1/2}}{V^{5/2}} = B_u$$

$$\text{Therefore } e_p = \left[\frac{B_u}{B_p}\right]^2$$

Admiral Taylor has plotted the B_u data in his Figure 208 and this is a much more valuable lot of information for it is generally required to choose a propeller for a given job rather than to determine what a given propeller will do.

It will be noticed from Admiral Taylor's curves that as the values of B_u increase, the maximum efficiency obtainable decreases and the correct pitch ratio to obtain this maximum efficiency also decreases. Capt. Lybrand P. Smith, U. S. N., in his article in the November, 1935, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS (5) brings this point out admirably and

shows how there is one best value of $\delta = \frac{nd}{V}$ and one pitch ratio which should be used to obtain the highest possible efficiency for a given B_u. Any deviation from this maximum efficiency curve should be avoided if possible.

To consider the phenomena of cavitation, there is apparently nothing better available than the formula devised by Capt. R. F. Eggert (C. C.), U. S. N. (6).

During the writer's period of experience in propeller design in the Bureau of Engineering of the United States Navy, he applied Capt. Eggert's formula to many propellers for difficult jobs, subsequently checking by running a model propeller at the United States Experimental Model Basin and, invariably, the Eggert criterion was dependable, *provided the propeller was not too great a distance from the maximum efficiency curve.*

As a propeller with a given pitch ratio was run at a value of B_u greater or less than that indicated for best efficiency, the cavitating speed became a smaller and smaller percentage of that indicated by the Eggert formula, being greatest when right on the maximum efficiency point.

This happened so invariably that the writer became fearful of predicting propeller performance when forced out of the area surrounding best efficiency. Not enough tests were obtained to ascertain just how far it was safe to go. This necessity for working in undesirable and questionable territory was, of course, always brought about by lack of clearance forcing the diameter of the wheel down, or by high speed line shafting due to many considerations.

It appears that this is a phase of propeller performance well worth investigating.

It should be borne in mind that Admiral Taylor has plotted only the characteristic curves of propellers having certain numbers of blades and certain blade shapes. He has also included correction curves for reasonable deviations from the shapes he uses as a basis.

In the same manner data could be obtained (and plotted) covering blades shaped for stream line flow, for non-conventional arrangements such as the Voith-Sneider, or any other such thing.

And from this data a design for a new proposition could be quickly and accurately obtained.

Good steam turbine designs are built up from good individual nozzle and blade designs. Good induction motor designs are built up from good slot and tooth designs. And good propeller design follows the same course.

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4. Edgar Buckingham—"Dimensional Analysis of Model Propeller Tests," Journal of A. S. N. E., Vol. 42, No. 2, May, 1926, pp. 147-198.

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SOME OBSERVATIONS CONCERNING METAL CRACKING IN NAVAL BOILERS.

By William C. Stewart, Member.*

During the past fifteen years there has been much published concerning the intercrystalline cracking of boiler steel. The volume of published information on this subject has directed thought toward caustic embrittlement whenever cracks are encountered in boiler shells. While the author recognizes the importance of investigative work in this field, he considers other factors which are often responsible for boiler metal failures. Thus, he correlates causes according to their relative importance from his wide experience of phenomena observed and recorded on the subject. He presents his findings and conclusions in an available form.

A discussion of this kind is concerned primarily with failed material. Hence, it should not be construed that this is indicative of any widespread trouble in Naval boilers. On the contrary, considering the hundreds of boilers in service, those having come to the attention of the Station as a result of metal failures are relatively few, and for the most part obsolete.

Only one case of boiler plate cracking, which can be classified definitely as intercrystalline, has come to the attention of the Engineering Experiment Station during the past ten years, and in this instance the drum was removed from Service for other reasons than the embrittlement cracks which were found on subsequent examination. Regardless of these circumstances there seems to be a feeling in some quarters that intercrystalline cracks are particularly insidious and that their distinguishing characteristics may be overlooked.

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Embrittlement cracks tend to weaken the plates slowly. The cracks are confined to riveted seams and pads where leaks develop. Leaky seams serve as a warning to operators and if care is exercised a casualty may be avoided.

General Corrosion.

Some of the more important facts concerning the mechanism of corrosion in general will be reviewed before considering the subject of embrittlement. Iron, like other materials, tends to go into solution in water. This tendency of a solid to pass into solution is termed the solution pressure. At normal temperatures iron will not corrode unless moisture is present. In the absence of oxygen iron reacts with water to form black magnetic oxide of iron (Fe_3O_4) and hydrogen as the final products. This reaction continues until the formation of an oxide coating stops further action by keeping the water away from the surface of the metal. However, the presence of oxygen is necessary if the corrosion rate is to be appreciable in ordinary water. Corrosion in acid solution is more rapid than in either neutral or alkaline solution and the least rapid in the latter.

For ordinary metal surfaces the initial rate of corrosion is usually much greater than the rate after a short period of time. This decrease in the rate of corrosion is particularly prominent in alkaline solutions. An increase in the concentration of neutral salts results in an increase in corrosion rate up to a certain point. Further concentration causes the corrosion rate to decrease to a minimum, other factors remaining constant.

If corrosion in boilers proceeded so that the metal was uniformly attacked there would be little or no difficulty. Under these conditions the plate thickness would be gradually decreased with the drum becoming obsolete before the danger point was approached. Unfortunately corrosion attack in boilers is sometimes selective, resulting in pitting and cracking. Experience has indicated that dissolved oxygen is probably the most important single factor in the corrosion of boilers. Hence, mechanical or chemical processes for reducing the oxygen content of the water to a minimum should do much to prolong the life of boilers.

Caustic Embrittlement.

As previously mentioned, corrosion is not usually serious unless localized or selective. Embrittlement is essentially the result of selective chemical action on steel, the grain boundaries being the avenues of attack. Cracking occurs mainly in riveted seams or other highly stressed areas where conditions are favorable for the concentration of solution. This type of failure attends a condition of high caustic alkalinity in the boiler water. The intergranular nature of the cracks is very important for identifying the type of failure.

Information available to date shows that concentrations of caustic soda of 100,000 to 500,000 parts per million of solution are necessary to produce caustic embrittlement in boiler plate. Hence, in order to produce cracking it is necessary to concentrate the water locally several hundred fold. Obviously, conditions must be favorable for this to occur. A favorable condition is a slight seam leak where minute quantities of the solution enter into a clearance space either around the rivets or between the plates. Since the amount of solution entering at one time is small and since the steel is hot, evaporation takes place with the result that the salt is deposited and the vapor phase escapes through the leaky seam. This process is repeated so that finally there is a salt deposit which is wetted each time more solution enters the clearance space. This mechanism would account for intercrystalline cracking in boilers although the boiler water normally would not be embrittling. The fact that salt incrustations are frequently found on the outside of drums where leaks occur seems to confirm this opinion.

In order to demonstrate that boiler water can be concentrated sufficiently to produce embrittlement, W. C. Schroeder and his coworkers at the Eastern Experiment Station of the Bureau of Mines have constructed a bomb which is used to simulate conditions of a leaky seam in a boiler drum. This bomb is cylindrical in form, and provided with a plug valve. The plug is grooved circumferentially at midlength and slight clearance is provided both below and above the groove. Under temperature, the water leaks slightly by the lower end of the plug depositing in the groove.

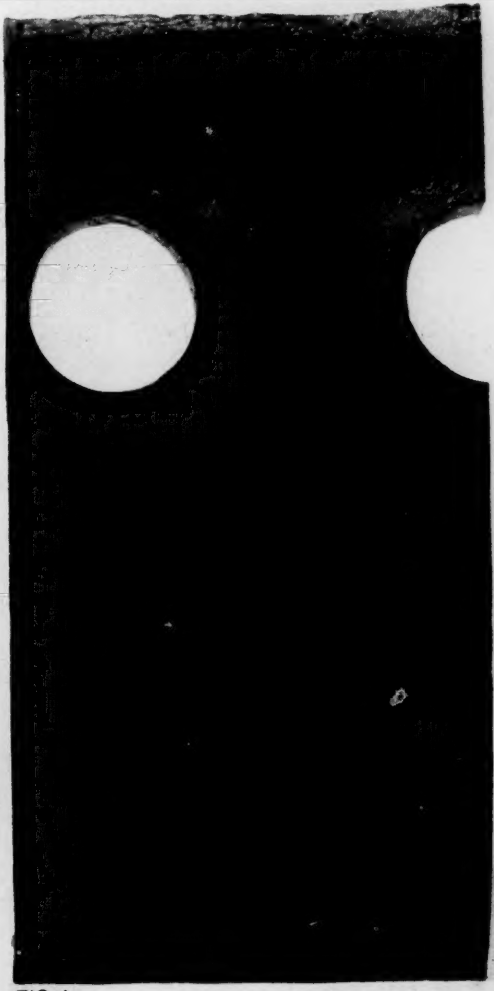

FIG. 1

FIG. 2 X 100

FIG. 3 X 300
UNETCHED

FIG. 4 X 300

SAME AS FIG. 3 - ETCHED

Here the solution evaporates, the vapor phase escaping through the clearance at the top of the plug leaving salt deposit in the groove. As more water comes in contact with the deposited salt the resulting solution is concentrated. This process is repeated until very concentrated caustic solutions are produced.

The application of this device has been extended so as to indicate when the composition of a boiler water is such that intergranular corrosion attack may occur. This is accomplished by connecting the bomb to the bottom of the boiler. The circulating water is allowed to escape past a strip of cold deformed steel contained in the bomb. Failure of the steel strip indicates that the boiler water is embrittling.

More Recent Views Regarding Caustic Embrittlement.

Continued investigation of the intercrystalline cracking of boiler steel by W. C. Schroeder and his co-workers has indicated that a broader viewpoint is necessary for explaining some of the more recent observations. It has been found that at 250 degrees C. very small amounts of sodium silicate in the sodium hydroxide greatly influence cracking. Additions of a number of other salts have a similar result while other chemicals such as sodium nitrate can be substituted for sodium hydroxide. However, laboratory experiments have failed to produce cracking in dilute caustic solutions. It has been shown that it is not necessary to have an applied load on the steel to produce cracking. The internal stresses resulting from cold work or other treatment seem to be sufficient. However, no intercrystalline cracking of annealed steel has been noted.

Figure 1 shows the outside surface of a boiler head plate. The surface shown was adjacent to the inside surface of the shell and doubling plate. The radiating cracks do not extend through the plate, there being a narrow band of sound metal at the water side of the plate. Consequently, these cracks could not have been observed before removal of the rivets and separation of the plates. The cracks in the head plate section reveal intercrystalline characteristics. A slight leakage must have existed in this area resulting in a concentration of caustic solution between the plates. Photomicrographs of these cracks are shown in Figures 2, 3 and 4.

Boiler Rivets.

Boiler rivets may also fail by intercrystalline corrosion. An indication of embrittlement in rivets is the cracking off of heads. However, rivet heads may crack off from other causes such as improper driving, excessive hardness and improper material. Microscopic examination is necessary in order to determine whether failure was the result of embrittlement and this should be supplemented with information as to boiler pressure, temperature, feed water treatment and any condition of local leakage. Cracks in rivets are particularly difficult of interpretation, on account of grain distortion as a result of driving. Obviously, when rivet heads crack off the rivet holes and adjacent areas should be examined for evidence of cracks.

Rivet failures in Naval boilers have been confined particularly to the rivets in pads. Only one case of possible embrittlement has come to the attention of the Station.

Corrosion Fatigue.

In 1926, McAdam while investigating the cooling effect of a stream of water when applied to the critical area of rotating cantilever endurance specimens found that under these conditions steel specimens failed under stresses far below the ordinary endurance limit. This led to a comprehensive series of experiments to determine the influence of cyclic stress on corrosion. The first part of this general investigation was concerned with the effect of cyclic stress at relatively high frequency simultaneous with corrosion. The combined effects of these factors as producing failure in metals under specific conditions of test was called "Corrosion Fatigue." In these experiments no attempt was made to differentiate the process and thus determine the proportion of damage caused by cyclic stress and corrosion.

Results of this investigation for a large number of ferrous and non-ferrous alloys were published in a series of papers between 1926 and 1928. These papers present a systematic study of the corrosion fatigue process as affected by chemical composition, heat treatment and cold working.

The following generalization may be made concerning the corrosion-fatigue process:

(1) Corrosion is accelerated by cyclic stress resulting in pits which are sharper and deeper than those produced by stressless corrosion.

(2) Transverse cracks or fissures develop at these pits as a result of local stress concentration. Thus, the actual stress at the bottom of the cracks increases with a corresponding increase in the rate of attack. This process continues until the stress at the bottom of the cracks exceeds the endurance limit of the material when failure occurs. These transverse cracks are regarded as chemically formed notches.

Stress-cycle graphs plotted on a semi-logarithmic scale for specimens tested in air and as free as possible from corrosion approach a horizontal asymptote. For steels the graph is usually horizontal at 10^7 cycles. This limiting stress is known as the endurance limit and is numerically equal to one-half the endurance range for complete reversal of stress per cycle. The endurance limit is an adherent property of the material in the same way as is the tensile strength.

Likewise, stress-cycle graphs for corrosion fatigue tests at relatively high cycle frequency become asymptotic if plotted on a semi-logarithmic scale. This similarity between the endurance and corrosion fatigue stress cycle graphs probably caused McAdam to "coin" the term "corrosion fatigue limit." Apparently, the latter was considered as a descriptive term and not intended to imply a definite limit as in the case of the endurance limit. The writer knows as a result of his association with McAdam during this period that he repeatedly cautioned against using "corrosion fatigue limits" in design, thus, showing that he did not regard these so-called limits as intrinsic properties of the material but useful as an index for comparing materials and interpreting metal failures.

It is obvious that "corrosion fatigue limits" do not represent a limiting condition of stress for it is well known that metals are damaged as a result of stressless corrosion.

Endurance and corrosion fatigue graphs for three steels are shown in Figure 5. These graphs are plotted on a semi-logarithmic scale. Ordinates denote stress in pounds per square inch and abscissas cycles. Each set of graphs has its own scale of ordinates;

the scale of abscissas is the same for all graphs. The stress-cycle relationship for specimens tested in air is represented by the solid line curves. The broken line curves represent the stress cycle re-

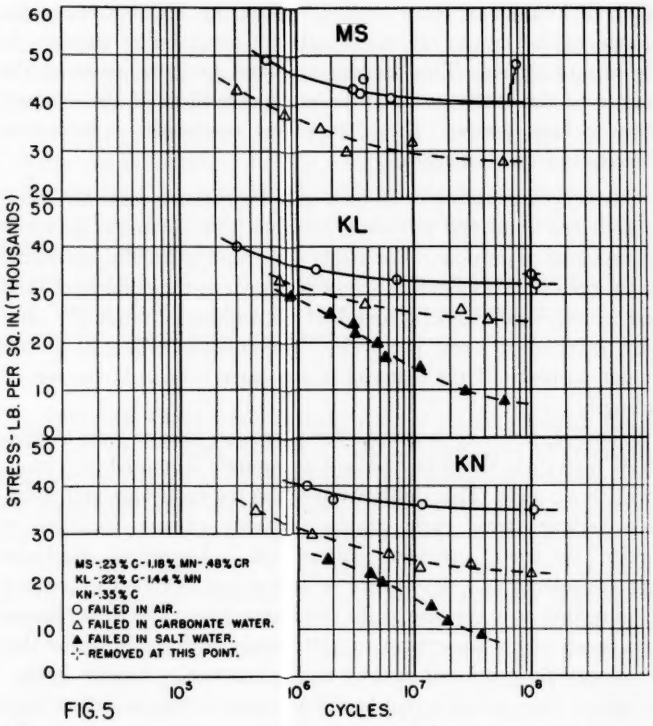
MS
KL
KN
STRESS - LB. PER SQ. IN. (THOUSANDS)
CYCLES.
MS - .23 % C - 1.18 % MN - .48 % CR
KL - .22 % C - 1.44 % MN
KN - .35 % C
○ FAILED IN AIR.
△ FAILED IN CARBONATE WATER.
▲ FAILED IN SALT WATER.
-⊹- REMOVED AT THIS POINT.

FIG. 5

lationship for specimens under cyclic stress simultaneous with corrosion. The open triangles represent tests made in fresh water while the solid triangles represent tests made in Severn River water, a brackish estuary water. These graphs are presented as being typical for steels of this type. The chemical and physical properties for the steels designated MS, KL and KN of Figure 5 are given in Tables I and II.

Fatigue Corrosion.

The second part of the investigation by McAdam consisted in determining the interrelationship of stress, time, number of cycles and corrosion environment. In this work the corrosion fatigue process was considered as consisting of two stages, an initial or pitting stage and a final or fatigue stage. In the first stage the specimens, while exposed to a stream of water, were subjected to a constant stress range at a constant frequency for a definite number of cycles. Since the frequency was constant throughout the test a given number of cycles was equivalent to a definite corrosion time. The second or fatigue stage consisted in determining the fatigue limit for the previously corroded specimens. The fatigue limit obtained from the second stage test is called the resultant fatigue limit.

Total damage resulting from the corrosion or first stage test is determined by the amount the resultant fatigue limit is inferior to the original fatigue limit. This total damage is made up of the damage due to stressless corrosion plus the excess of net damage due to cyclic stress. The influence of net damage is measured by net damage rather than total damage.

For the purpose of this paper it is only necessary to consider the stress corrosion process from the viewpoint of ordinary and low alloy steels. Steels of this type, having a considerable range of chemical composition, were included in the experiments. Some of the steels were tested in more than one condition of heat treatment.

In order to understand the procedure which McAdam devised for studying the several variables, it is necessary to consider the so-called Type 5 graphs. In these diagrams abscissas represent corrosion times and ordinates represent resultant fatigue limits. These diagrams are used as a basis for deriving more complex graphs. Figure 6 shows a series of type 5 diagrams for 3½ per cent nickel steel in the quenched and drawn condition. It will be noted that the frequency ranges from 1.5 to 10,000 RPM; each series of graphs representing some one cycle frequency. The form of the graphs of Figure 6 are characteristic of steels but not necessarily of other alloys. Each plotted point in these figures represents the

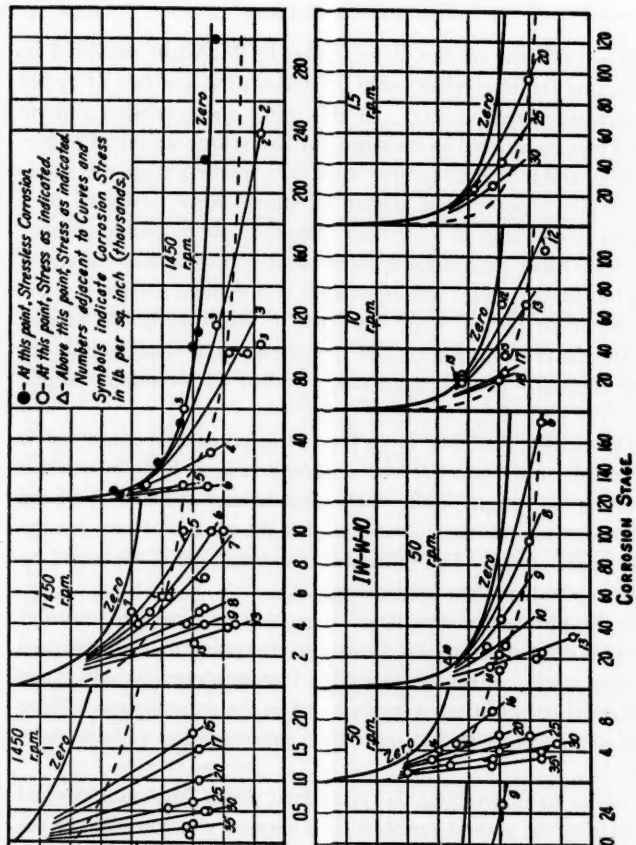

FIG. 6

DIAGRAMS FOR NICKEL STEEL.

result obtained for a single specimen. The resultant fatigue limit represented by the ordinate of the point is obtained by extrapolation, the length of the normal from the endurance graph to the point being equivalent to the lowering of the fatigue limit. The graphs including those for zero stress have a common origin; namely, the endurance limit. The higher the corrosion stress the steeper the course of the graphs. However, the curvature for the corrosion stress graphs is similar to the curvature for the stressless corrosion graph. These graphs as they extend to the right have a continually decreasing slope. For this reason they have been designated retarded damage graphs.

Attention is called to the broken line curves shown for each series of graphs in Figure 6. These lines are drawn so that the ordinates will be a constant percentage of the corresponding ordinates of the stressless corrosion graph. For the graphs as drawn the ordinate for the broken line graph is 85 per cent of the ordinate of the stressless corrosion graph, thus representing 15 per cent net damage. Intersections of the corrosion stress graphs with the net damage graphs represent constant net damage. The latter may be defined a constant (percentage) lowering of the fatigue limit below the fatigue limit that would result in the same time for stressless corrosion. McAdam used the 15 per cent value as a convenient figure but other values could be used as well.

The type 5 graphs are used as a basis for the construction of other types of diagrams to represent the interrelationship of stress time and cycles in producing 15 per cent constant net damage. Two types of derived graphs have proved most useful, namely, diagrams illustrating the influence of corrosion stress on net damage (Type 11(a)), and diagrams illustrating the influence of total number of cycles and of cycle frequency on net damage (Type 10). The interrelationship of the three variables can be represented by the surface of a three dimensional model. For the purpose of this paper only the type 11(a) diagram will be considered. Those interested in the other type diagrams are referred to the original papers by McAdam.

Type 11(a) graphs, each representing one cycle frequency plotted to logarithmic co-ordinates are shown in Figure 7. In each

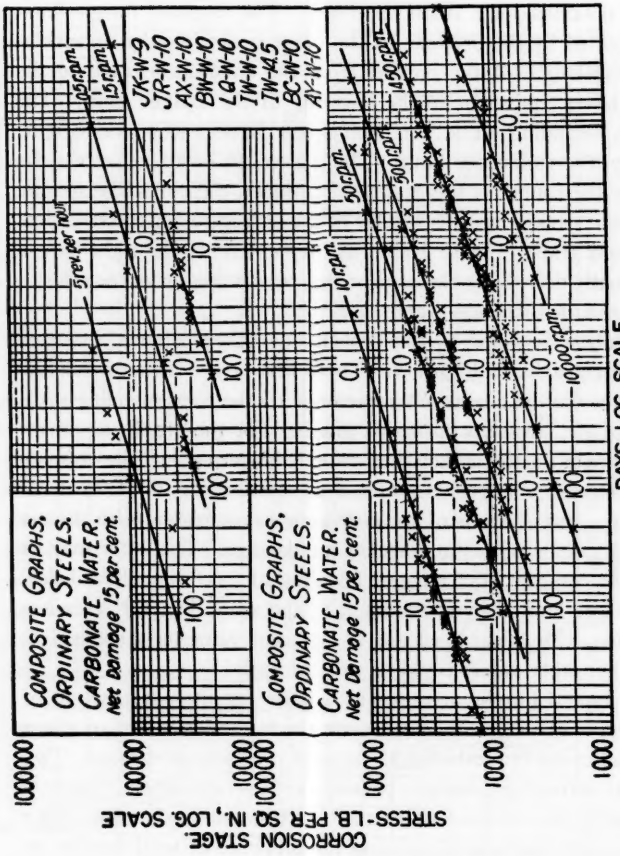

FIG. 7 TYPE II (A) DIAGRAMS - STEELS IN CARBONATE WATER.

division, the graphs have the same scale of ordinates but each graph has its own scale of abscissas which are indicated by numbers adjacent to the graphs. These graphs represent constant net damage of 15 per cent, abscissas measured from right to left represent corrosion times. Since the corrosion time for 15 per cent net damage varies inversely as the average rate of net damage for a total of 15 per cent net damage, abscissas measured in the opposite direction, left to right, represent rates of net damage.

The diagrams of Figure 7 represent composite graphs for ordinary steels. The steels include carbon steel, 3½ per cent nickel steels in the annealed and quenched and drawn conditions, several chromium-nickel steels in the quenched and drawn condition and a copper bearing steel. The tensile strength for this series ranges from approximately 90,000 to 157,000 pounds per square inch. Considering the wide range of composition and physical properties of the steels represented by the graphs, the scatter of experimental results is surprisingly small. Each composite graph representing a definite frequency closely approximates a straight line; the graphs are nearly parallel, the slopes showing a slight increase with increase in frequency. This logarithmic straight-line relationship means, that, (1) the influence of stress on corrosion pitting for ordinary and low alloy steels is practically independent of composition and physical properties; (2) that, the rate of net damage varies as a power of the corrosion stress. This relationship may be represented by an equation of the following type:

$$R = CS^N$$

where R represents the rate of net damage, S is the semi-range of cyclic stress, C is a constant and N is the exponent. The factor C depends on the corrosion resistance of the metal, corrosiveness of the solution and cycle frequency. The factor N is the cotangent of the slope of the graph and is practically constant for the steels considered. For steels in carbonate water with free access of air, N ranges from about 2.8 to 3.5. Under boiler conditions the numerical value for the exponent N would be approximately 3.

No one having had experience with failed outboard marine propeller shafts can doubt the damaging effect of corrosion simultaneously with repeated stress. Normally these shafts are protected by bronze sleeves placed about the shaft in sections. A paste consisting largely of red lead is used to seal the overlapping joints. Failure of this packing may allow water to penetrate and come in contact with the surface of the shaft. Under these conditions a circumferential band of corrosion pits develops in the region of the sleeve joint. These pits progress in an accelerated manner until the actual stress at the bottom of the corrosion cracks exceeds the endurance limit, when failure results.

FIG. 7 TYPE II (A) DIAGRAMS - STEELS IN CARBONATE WATER.

The mechanism of fatigue corrosion as related to the equation $R = CS^N$ has been used to explain the occurrence of transcrystalline cracks in boiler drums and headers. Objections to this procedure have been raised in some quarters. These objections are based mainly on the following considerations:

(1) The stress corrosion test data represent conditions of corrosion in air at normal temperature.

(2) The fatigue corrosion process depends on cyclic stress which does not necessarily obtain in boilers. Moreover, the stresses in boiler drums range between zero and a maximum whereas the stress-corrosion graphs represent results for alternating stresses.

(3) Boiler waters are much less corrosive than the carbonate water with free access of air used in the experiments.

Stressless Corrosion Tests in Boiler Environment.

Recently, some experiments have been made at the Engineering Experiment Station to determine the effect of stressless corrosion under boiler conditions. In these experiments fatigue specimens of the same type as those used in the original experiments were placed in a boiler drum so as to be entirely submerged in the circulating water. Comparison tests for specimens of the same materials were made in carbonate water. The maximum period of test for the boiler specimens was approximately 1950 days, other tests were continued for approximately 1550 days. The boiler was operated intermittently and when in use the temperature was 400 to 425 degress F. No provision was made for removing oxygen from the feed water. The feed water was treated in accordance with a modified form of the standard Navy feed water treatment.

In making the tests, specimens were taken from the boiler drum or carbonate water stream at various intervals and the resultant fatigue limits determined after the method previously described. Results of test for a 0.35 per cent carbon steel, a manganese steel and a low chromium-manganese steel are shown in Figure 8. Similar results of test for two types of nickel steel are shown in Figure 9. For the 3¼ per cent nickel, steel material designated MV, tests were made in two conditions of heat treatment. The origin

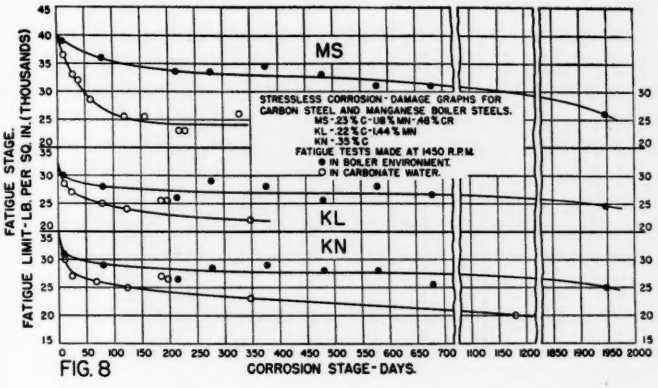
MS
KL
KN
STRESSLESS CORROSION-DAMAGE GRAPHS FOR CARBON STEEL AND MANGANESE BOILER STEELS.
MS-·23% C-·18% MN-·48% CR
KL-·22% C-1·44% MN
KN-·33% C
FATIGUE TESTS MADE AT 1450 R.P.M.
● IN BOILER ENVIRONMENT.
○ IN CARBONATE WATER.
FATIGUE STAGE.
FATIGUE LIMIT-LB. PER SQ. IN. (THOUSANDS)
CORROSION STAGE-DAYS.

FIG. 8

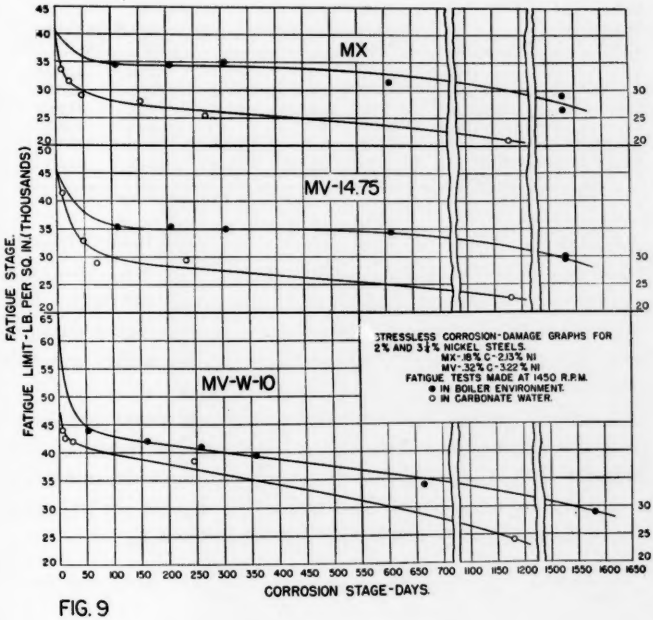
MX
MV-14.75
MV-W-10
STRESSLESS CORROSION-DAMAGE GRAPHS FOR 2% AND 3½% NICKEL STEELS.
MX-·19% C-2·13% NI
MV-·32% C-3·22% NI
FATIGUE TESTS MADE AT 1450 R.P.M.
● IN BOILER ENVIRONMENT.
○ IN CARBONATE WATER.
FATIGUE STAGE.
FATIGUE LIMIT-LB. PER SQ. IN. (THOUSANDS)
CORROSION STAGE-DAYS.

FIG. 9

of the graphs represents the endurance limit of the metal as determined for uncorroded specimens, ordinates represent resultant fatigue limits and abscissas corrosion time. The chemical and physical properties for the several steels are given in Tables I and II, respectively.

TABLE I
Chemical Composition

Material designation	C	S	P	Mn	Si	Ni	Cr
	Per cent						
MS	0.23	0.016	0.028	1.18	0.67	<.05	0.48
KL	0.22	0.028	0.007	1.44	0.01	0.11	.017
KN	0.35	0.030	0.011	0.81	0.21	0.02	.008
MX	0.18	0.018	0.018	0.63	0.17	2.13	-
MV	0.32	0.014	0.034	0.48	0.14	3.22	-

TABLE II
Physical Properties

Material Designation	Condition	Tensile Strength	Proof *Stress	Elongation in 2 inches	Reduction of area	Endurance Limit-Rotating Cantilever
		Lb/sq. inch		Per cent		Lb/sq.inch
MS	As received	96900	40500	26.7	64.9	40000
KL	As received	70000	34900	36.5	73.3	32000
KN	As received	82000	36800	30.0	53.8	35000
MX	As received	74300	49000	31.9	53.4	40500
MV-14.75	Annealed	84400	52700	28.7	48.6	45500
MV-W-10	Quenched and drawn	111200	94800	23.3	64.8	65000

As would be expected these graphs show that boiler environment is not as damaging as stressless corrosion in air. However, the graphs show a downward slope as they extend to the right. This downward slope is much more pronounced for the higher strength steels. The total damage is also greater for the higher strength steels.

The method of test just described provides a means for determining the approximate intensity of the corrosion factor C. Of course the time required for such type of test would be much less than for the tests described. The greater the slope of the stressless corrosion graph, the greater is the corrosion factor C.

* Stress necessary to produce .0001 inch per inch permanent deformation after removal of the load.

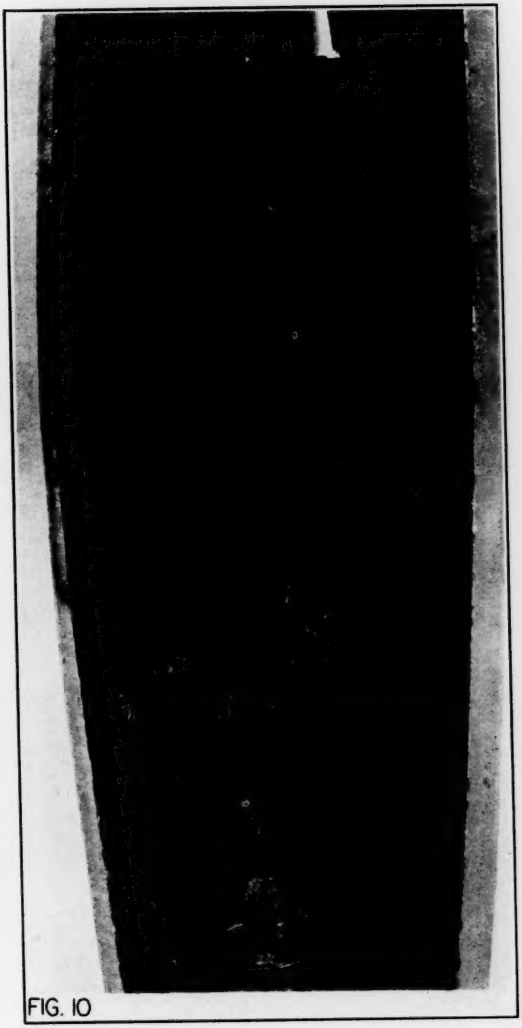

FIG. 10

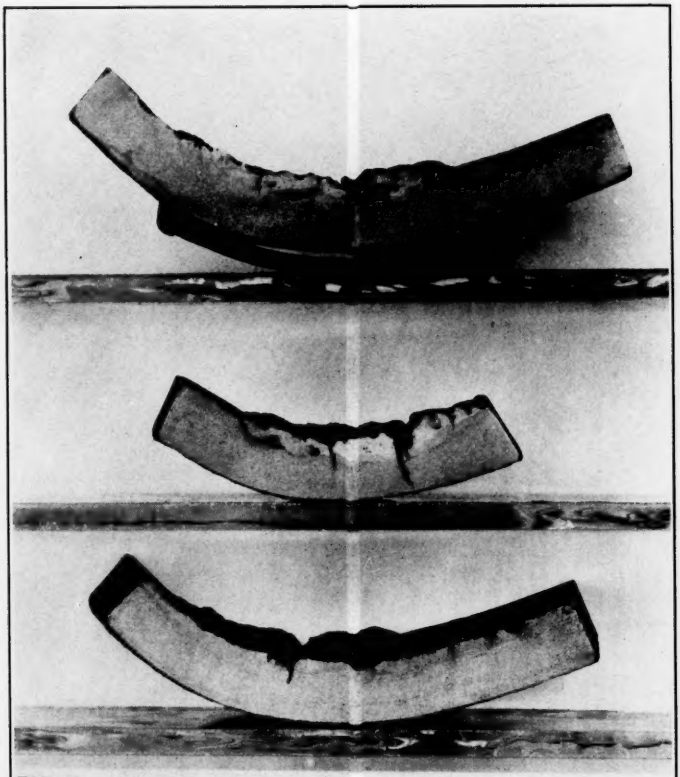

FIG. 11

FIG. 12 X 100

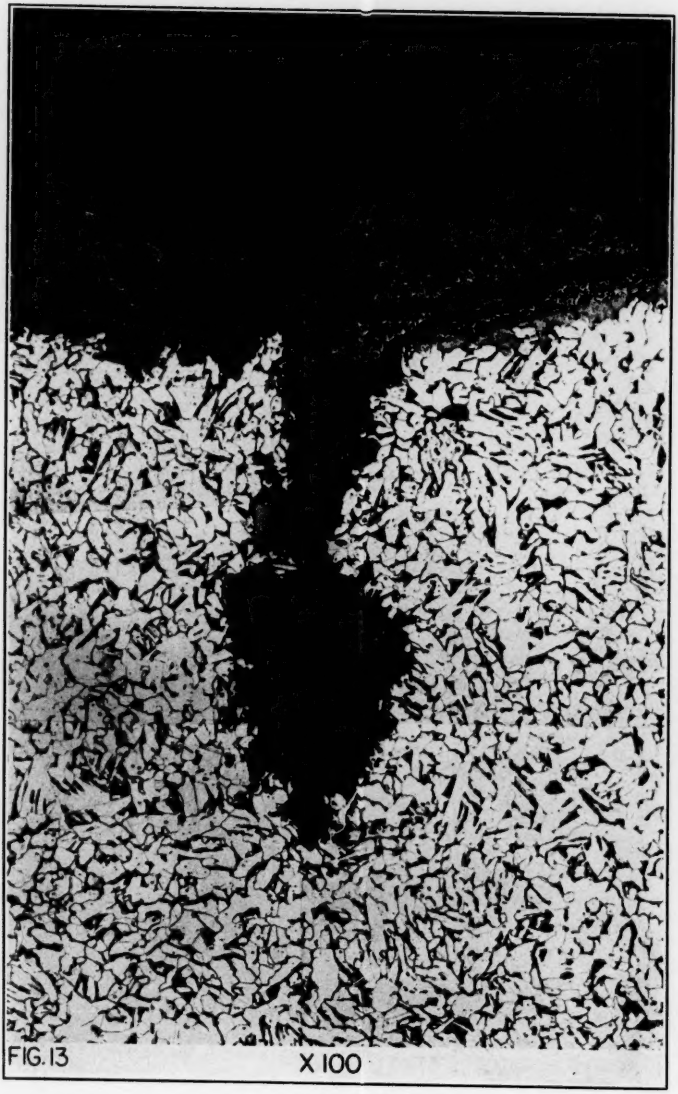

FIG. 13 X 100

FIG. 14

FIG. 15 X 100

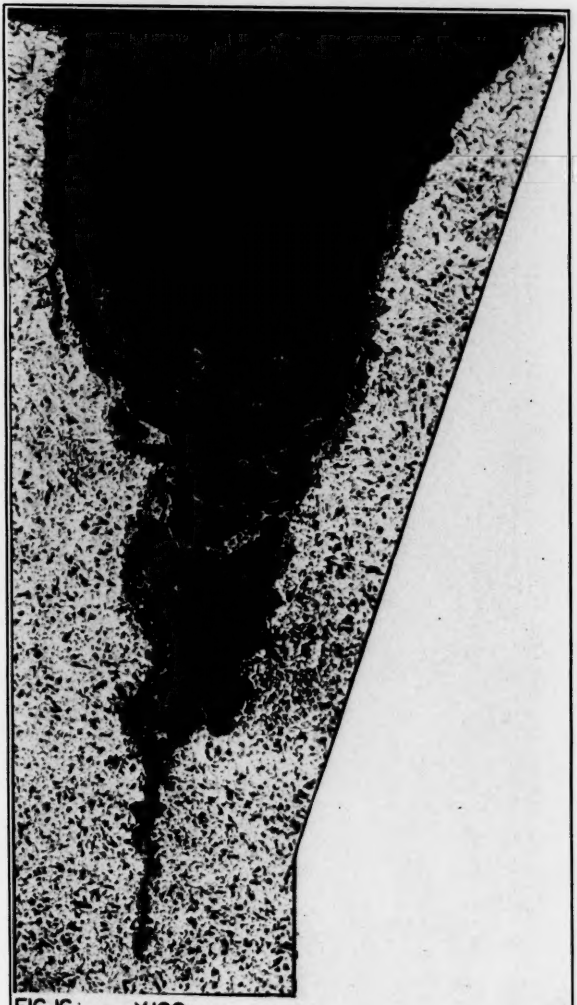

FIG. 16 X100

FIG. 17 X100

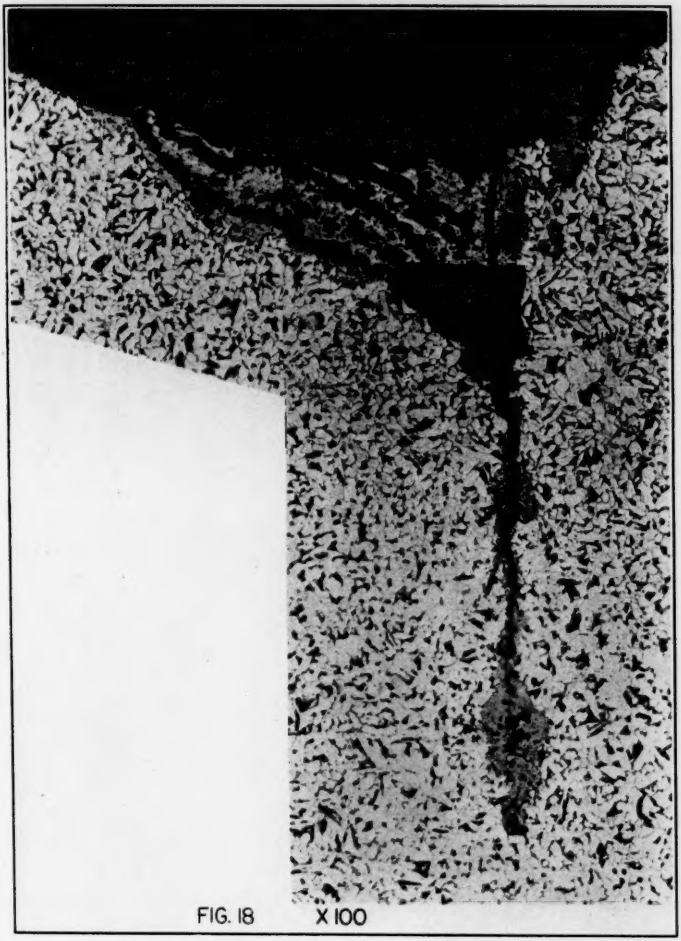

FIG. 18 X 100

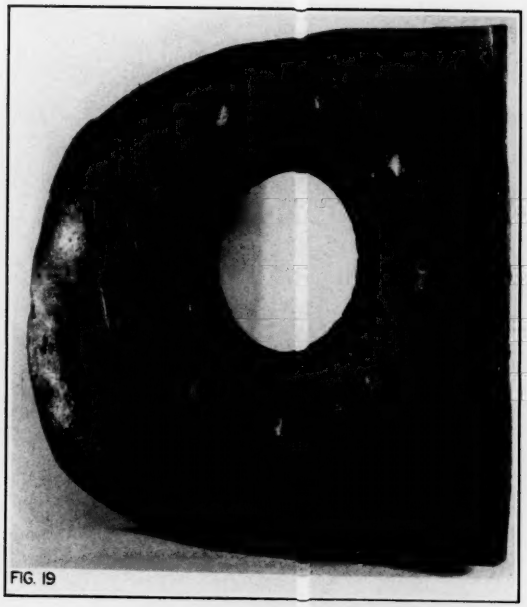

FIG. 19

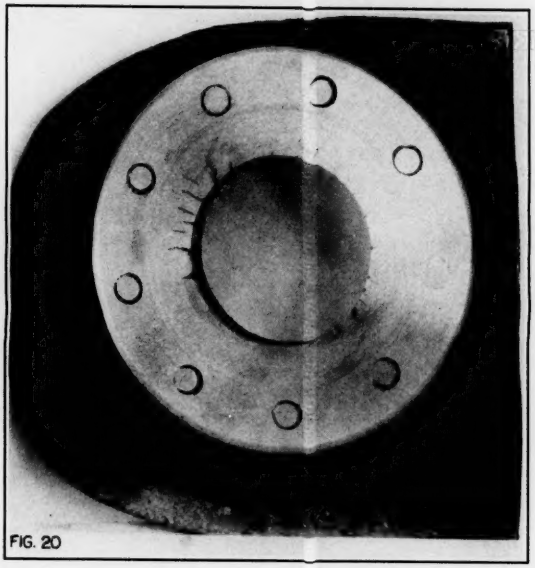

FIG. 20

A stress cycle occurs each time a boiler is lighted off. Stresses occasioned by variable pressure are also of a cyclic nature. Repeated heatings and coolings occasioned by feed water passing over hot metal intermittently would introduce stresses to the surface of the metal. Attention was called to this condition by A. E. White in a paper, " Changes in a High-Pressure Drum to Eliminate Recurrence of Cracks Due to Corrosion Fatigue ", presented at the December 1938 Meeting of the A. S. M. E. In boilers cyclic stress ranges between zero and a maximum whereas the experimentally determined graphs represent results for complete stress reversal each cycle. A stress range from zero to a maximum is equivalent to one-half the corresponding range for alternating stress.

Boiler Metal Failures in Service.

Figure 10 shows a deep corrosion crack about 8½ inches located in the knuckle of a boiler drum head. The surface reveals a rough condition which resulted from weld deposits having been laid down in an attempt to retard corrosion attack. The depth of these cracks is illustrated in the cross sectional surfaces shown in Figure 11. Photomicrographs showing the characteristics of representative cracks are presented in Figures 12 and 13. Figure 12 shows the extremities of a forked fatigue corrosion crack while Figure 13 shows a relatively short, broad fissure with a bulbous accumulation of corrosion products at the end. This latter photomicrograph represents the extreme end of one of the cracks. The cracks are lined with corrosion products, water reaching the advancing edge of the crack through the main channels. It will be noted that these cracks are transcrystalline as distinguished from embrittlement cracks which are intercrystalline.

Particularly serious fatigue corrosion cracks are illustrated for the cross sectional surfaces illustrated in Figure 14. These sections were taken from the knuckle area of a boiler front drum head. The notched appearance of the water side of the sections is shown in the two lower figures. Photomicrographs showing two of these cracks are shown in Figures 15 and 16. The advancing edge of these cracks is very sharp resulting in high stress concentration.

Photomicrographs showing similar cracks in another drum head are shown in Figures 17 and 18. These photomicrographs clearly

illustrate how the corrosion cracks extend from the bottom of corrosion pits.

Corrosion cracks around the inlet bore of a feed water inlet flange are shown in Figure 19. The condition of the flange after removing .150 inch from the face and .100 inch from the diameter of the bore by machining is illustrated in Figure 20.

CONCLUSIONS.

During the last few years there has been a tendency to use continually higher boiler pressures. This has resulted in a tendency to increase the working pressure and thereby reduce weight. Welded boiler construction has eliminated lapped seams which were favorable for the concentration of caustic soda solution. This type of construction also has eliminated rivet holes except in cases where riveted steam inlet and water gage pads are used. High initial stresses may be introduced into boiler plate by careless fabrication. This condition is practically eliminated in welded drums as a result of the final stress relieving heat treatment. Notwithstanding these improvements, the drum knuckles and feed water inlets are regions of stress concentration and excessive stress concentration may result from improper design.

Illustrations presented show that fatigue corrosion cracks develop in regions of high stress concentration, namely, the knuckles of the head flanges and the feed water inlet pads. Since the rate of net damage varies as approximately the cube of the stress, the latter should be kept as low as practicable. The corrosion factor can best be reduced to a minimum by keeping the oxygen content of the water as low as possible. As regards present feed water treatment both boiler drums and tubes have been unusually clean and free of scale for the past several years.

The writer sees no advantage in a continuation of the controversy as to whether or not intercrystalline cracking is a special case of corrosion fatigue. Both phenomena have their own distinguishing characteristics and if the future can be forecast by the past, the Navy should have little trouble as a result of intercrystalline cracking in boiler drums.

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A SHORT HISTORY OF THE NAVAL USE OF FUEL OIL.

PART VI.

By Lieutenant Commander James E. Hamilton, U. S. Navy, Member.*

This part of the author's history covers the period from 1920 to 1934 under the subhead "Source." He discusses the natural sources of petroleum and possible substitute or replacement materials; their location, control, and Naval value. The Navy's oil reserves play a prominent and dramatic part in this period. The author presents an authoritative and unbiased picture of the events which arose from attempts to use the Navy's oil fields for immediate profit. He suggests the importance of technical information which was derived concerning the fields during the period of partial operation. Lessons in the future handling of the reserves may be drawn from his discussion.

SOURCE.

At no time had any real effort been made to obtain a current Government or Naval source for fuel oil required by the fleet. The few and scattered suggestions to this end had never been definitely accepted officially. Secretary Josephus Daniels, in waging his successful fight for the retention of the Naval Petroleum Reserves, had visualized the fighting ships using oil produced from the Navy's wells in its own fields and probably included in his picture the Naval topping plant which had been recommended by the 1916 Fuel Oil Board. However, this picture was someplace in the future.

No evidence was uncovered in the author's search of the records to show that the Oil Industry had openly opposed the Naval refinery idea. The evident opposition came from the Navy itself so there was really no opportunity to discover the attitude of Congress and the industry to the proposition.

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The commercial source continued to supply the Navy as it had from the very beginning and still does. The extensive construction of both naval and merchant ship tonnage which was undertaken as a portion of the national effort to win the war, by 1920 threw an unusually heavy demand on the oil industry for an adequate supply of fuel oil. This demand, coupled with the beginning of the phenomenal post-war growth in the use of the automobile, led to a vigorous exploratory campaign to find new sources of petroleum.

The result was the discovery of many prolific fields which were rapidly exploited, without restriction. All regions contributed to a flood of oil which started in 1921 and ran for three years. This flood of oil, rapidly draining the most valuable of the natural resources of the United States, made the discovery of the fact that the future supply of petroleum from Mexico would fall far below the expectations of the promoters a matter of small importance. The decline in Mexican production began in 1922.

The loss of Mexican oil was serious only as it upset the prognostications of the experts who had seen our southern neighbor as the future major source of our Navy's fuel oil. It threw us back sharply on our own resources and the fields of California, Texas, and Oklahoma again became the Navy's evident source of future supply.

NAVAL PETROLEUM RESERVES.

The post-war period was one of great importance to the Naval Petroleum Reserves. They passed through a peculiar cycle which brought them very prominently into the public eye.

The legislative effort to settle the public lands question, which affected the Naval Petroleum Reserves, lagged throughout the war only because of the greater importance to the nation of other questions. The people who had been attempting to grab the public lands tried, even during the war, to get Congress to present the lands to them. However, it was not until 1920 that the Public Lands Leasing Law was finally approved, thereby settling the matter as far as legislation was concerned.

The law provided for leasing of the Public Lands for oil production, but restricted leases on the Naval Petroleum Reserves to

wells which were already producing with sufficient land around each one to permit its operation. Also the President was authorized to lease the remainder of the claims on which producing wells were located, on any terms which he considered proper.

As there were no producing wells on Reserves Number 1 and 3, it appeared that these two could be saved intact. Reserve Number 2, consisting of 30,181 acres, was pretty thoroughly drilled up. When all of the claims to this reserve were finally settled, 20,115 acres were patented to private interests and all of the remainder which was oil bearing was leased under the law of 1920. The Navy collected royalties from oil and gas production on leased lands; but no benefit accrued to the Navy from these collections since they were all turned in to the Treasury as miscellaneous receipts. There was no other advantage to the government from these arrangements and the value of Reserve Number 2 as a Naval Petroleum Reserve was nil.

Thus 1920 found the Navy with two petroleum reserves which were practically intact and, although not definitely proved, were almost unquestionably oil bearing. The fight was not finished, however. The oil industry wanted oil and it wanted the oil in the Naval Petroleum Reserves. During the remainder of Josephus Daniels' term as Secretary of the Navy, efforts were made to get leases to these lands but Daniels had made too great an effort, largely on the insistence of Rear Admiral Griffin, the Chief of the Bureau of Steam Engineering, to hold the reserves. Furthermore he seems to have changed his earlier views that the Navy should itself produce oil from the reserves to fill its current needs and now felt that the oil in the reserves should be held in place in underground storage until the need for it should become acute. Under Griffin and Daniels the reserves were safe.

The story of the leasing of the Naval Petroleum Reserves has been told many times. The Senate, through a special committee, thoroughly investigated the details, and the courts reviewed the matter and found that the major leases had been fraudulent. Only one man, ex-Secretary of the Interior Albert A. Fall, was found criminally guilty. Altogether, a sordid set of circumstances was exposed but in all of the investigations the facts were not definitely proved nor can it be expected that they ever will be.

In his "The United States Oil Policy," John Ise devotes a chapter to the reserve scandal. His assumption on the testimony adduced is that the entire affair was rotten from top to bottom except that he very definitely absolves Naval Officers from having had any part in the fraud.

It appears throughout an investigation of the Navy and its fuel oil problems, that at no time has any commercial interest been willing to put the Navy's, and hence the Nation's, interest ahead of a possible excessive (not fair) profit. It is almost certain that the reserves would have been the prey of the oil industry until someone got them. It is probably fortunate that the first successful attack was made with so little finesse that its eventual defeat was inevitable. The popular sentiment which arose as a result of the Teapot Dome and Elk Hills publicity should serve to protect the reserves for years to come. Of all the elements in the oil industry, many would have accepted conditions and permitted the Navy to keep its reserves. Others would have gained the reserves in such a clever and apparently open way that the courts could have found no fault with the methods employed; and the exploitation of the reserves would have followed with complete legal sanction.

Fortunately the crude methods of perverted witnesses and not even subtle bribery were practiced. Very little real damage was done and as will be shown certain gains were made.

The conditions, as they existed at the time of the change of administrations, on March 4th, 1921, were as follows: Reserve Number 1 and 3 were in the Navy's possession, practically intact. There still existed certain unsettled mining claims which were being cleared up. Under the act of June 4, 1920, the Secretary of the Navy was "directed" to take possession of the naval reserves, "to conserve, develop, use, and operate the same." In the Navy Department, the Bureau of Steam Engineering, as the bureau of major interest, acted as general adviser to the Secretary as to matters of policy. The definite policy was established that the oil in the reserves should be conserved in the ground and should not be produced until other sources of normal supply failed.

On May 31, 1921, even in the face of the above quoted provision of an act of Congress, the President, by executive order, trans-

ferred the Naval Petroleum Reserves to the custody of the Secretary of the Interior. The excuse is one which is now used for many suggestions for reorganization of the executive branch of the Federal Government. It was that the Department of the Interior was better fitted to take care of the reserves and that, since it already handled the public lands, the transfer would do away with duplications of various kinds in the two departments.

Admiral Griffin opposed the transfer and obtained the support of the Assistant Secretary of the Navy, Theodore Roosevelt, but to no avail. Admiral Griffin wrote a protest to the President and fought for a provision in the transfer order whereby any leases contemplated on the reserves would necessarily be referred to the Navy Department before consummation. A thorough search of the files of the Navy Department indicates that this letter was never sent to the President by the Secretary of the Navy.

Charges have been made that the entire affair had been planned even before the 1920 election; and that the plan included the appointment of Fall as Secretary of the Interior as the first step and the transfer of the custody of the reserves as the second. Such charges have not and cannot be proved. Whether long planned or not, those were the actual events. It was charged that Secretary Fall wrote the transfer order but Secretary of the Navy Edwin Denby testified before the Senate investigation committee that he had proposed the transfer for the reasons given above.

In the meantime, the exploiting interests had taken steps to assure themselves of the value of the reserves. The estimates which were obtained for them were conservative in the case of Elk Hills, Reserve Number 1 and excessive for Teapot Dome, Reserve Number 3. It appears very probable in the light of present knowledge, that full exploitation of Teapot Dome would have resulted in a net loss to the Mammoth Oil Company, because the prolific production of the sister field to the north, which was used as a guide, did not continue southward into Teapot Dome.

In order that reasons could be given for operating the reserves, the drainage propagandists opened up their attack. Expert geologists of excellent standing declared that operations in producing oil from lands in private hands on the borders of the reserves and on

section 36, held in fee by a state grant by the Standard Oil Company of California and located almost in the productive center of Elk Hills, would drain all of the oil out of the reserves. It was predicted that when and if the reserves were eventually needed, it would be found that they had been drained dry or, at best, very much depleted. In that case the Navy and the government would get not a barrel nor a cent for all of the oil in the reserves, and would continue in a sense of false security in believing that the Navy had a safe reserve of fuel oil to use in any eventuality. The only way to remedy this situation was to drill wells on government land, offsetting each well located near the reserve boundaries, and produce the Navy's oil faster than it could be produced by the draining wells. The fuel oil equivalent of this crude oil could then be stored in sight in above ground storage; and the Government could derive a small, but otherwise lost, income from the royalties. Rear Admiral J. K. Robison, U. S. Navy, who succeeded Admiral Griffin as Engineer in Chief, was convinced, honestly and thoroughly, that the drainage theory was correct; and he favored operating the reserves immediately. He wanted the Navy and the government to get the greatest possible advantage out of its ownership of these very valuable oil properties, which appeard to be slowly slipping away. Admiral Robison was strongly influenced by two other considerations. The international situation in the Orient was none too happy; and many people felt that there was a very definite probability of war in that quarter within probably three years. Also he agreed with the 1916 Fuel Oil Board and the Navy General Board that the Navy's above-ground fuel oil storage facilities were sadly deficient; and, as described in the following section, Congress refused every request for funds to permit remedying the deficiency. The exploiters showed the way for the Navy to obtain this sorely needed tankage.

At this point the stage was set and the Navy's interests were very much in jeopardy. However, the next step saved the day. If, after reaching the conclusion that to prevent drainage the reserves must be brought into production and operated, the Secretary of the Navy had advertised for bids which would permit free competition, and had accepted that bid which appeared to be most

advantageous to the government, the reserves would have ceased to exist as such. The Navy and the Treasury would have gained an immediate but temporary advantage in fuel oil supply and cash income; but it would have been at the expense of assurance against a serious shortage of fuel oil supply at some future critical time.

Instead, because of fraud as many charge, or because of military necessity as Admiral Robison claimed, or because of greater advantage to the Government as Mr. Fall contended, the reserves were secretly leased without competition. The events were as follows:

On April 7, 1922, all of the land in Teapot Dome was leased to the Mammoth Oil Company which was controlled by Mr. Harry Sinclair. As compensation for the lease, the Government was to receive:

(*a*) Clear title to the reserve, Mr. Sinclair having secured quit-claims from all mineral claimants as one of the conditions of signing the lease.

(*b*) A pipe line was to be laid from the Salt Creek field in Wyoming to Kansas City, where it would tie in with the trunk pipe line systems which extended to the Gulf Coast. This would provide an outlet to salt water not only for the Navy's "own" oil from Teapot Dome, but would also make the general Wyoming production available for use on the seacoast in the event of any serious curtailment from other sources.

(*c*) The Navy would receive a royalty which ranged from 12½ per cent, which represented about what could be expected in the average commercial oil land leasing deal at the time, all the way up to 50 per cent, which was indeed a generous royalty rate. The royalty scale varied with the size (rate of production) of the wells which were to be drilled and put on production. Actually, while the lease was in effect, the average royalty which the Navy received from Teapot Dome was between 16 and 17 per cent.

The royalty oil which the Navy was to receive was crude oil, that is, petroleum as produced at the well. This oil could not be used as fuel oil, nor as gasoline nor lubricating oil, without refinery treatment. Since the Navy owned no refinery, the only value which the oil had to it was its sale or exchange value. If the

oil was sold, the law required that the proceeds from the sale must be turned into the Treasury as miscellaneous receipts; and the Navy would receive no benefit from the transaction, since it was unlikely that a comparatively small increase in the income of the Federal Government could have been reflected in an increase in appropriations for Naval purposes. Under the law, there was only one way in which the royalty oil could be disposed of to give the Navy any direct benefit. That way was to exchange it for fuel oil.

The current needs of the Navy for fuel oil for operating the fleet were adequately provided for by the annual appropriation acts; and with the money available there was no difficulty whatever in buying it in the open market. What the Navy wanted was fuel oil so stored and so located that it would be immediately available for use in the event of an emergency. But the Navy had no empty storage tanks in which to place the oil. The few it did have were in use and required for current distribution.

Who first proposed the solution finally adopted cannot be said. After considerable discussion and against some opposition, a liberal interpretation of the law was applied; and, under the terms of the lease, the royalty oil was to be exchanged for *stored* oil on the basis of one barrel of stored fuel oil for each three barrels of royalty crude oil. The stored oil was to be located at such points on the Atlantic coast as the Navy might direct. The net result would be that for each barrel of *crude* oil which the Navy owned stored in its natural underground reservoir, it was to receive from 1/24 to 1/6 of a barrel of fuel oil stored in tanks at strategic locations on the Atlantic coast. The arrangement certainly provided for the very essential emergency reserve fuel oil storage, ready for use, but it did so at the expense of the final and safe underground reserve.

In Elk Hills, events were somewhat different. Whereas Teapot Dome had been handed to the Mammoth Oil Company in one move, three were used in transferring the greatest of all reserves to the Pan American Oil Company which belonged to Mr. Doheny. The drainage scare had been withstood by Secretary Daniels until near the end of his last term of office. Then he finally capitulated to the extent of calling for bids for a lease to 21 well sites, which would offset wells adjoining and presumably draining Reserve

Number 1. Had the bids been received and acted on during his incumbency, it is possible that he might have negotiated a lease for operating the reserves but it would have involved a very narrow strip along one border only.

The lease was eventually made by Mr. Fall after the reserves had been transferred to his jurisdiction on July 12, 1921, and the award was made to Mr. Doheny's company. Apparently this transaction was made properly and in full accord with the law and gave Mr. Doheny the same rights which any other operator might have obtained by making a better offer to the Government in response to the invitation for bids.

A few months later, the Navy called for bids for exchange of stored fuel oil for royalty crude, similar to the arrangement made directly with Mammoth. Many companies who were given an opportunity, did not bid because they believed that such an exchange was not sanctioned by existing law. However, several bids were received in accordance with the Navy's proposal, but the award was made on an *alternate* bid submitted by Mr. Doheny. The claim was made that this alternative was more advantageous to the government; but the fact remains that no competition was permitted on the identical terms covered by the alternate bid. One presumably innocuous clause in the Doheny offer was to the effect that if the bid was accepted, his company, the Pan American Petroleum and Transport Company, was to be given preferential rights to lease the remainder of Reserve Number 1. This apparently meant nothing more than that if at some time in the future the Navy should for some reason decide that operations in Elk Hills must be extended, Pan American would have the right to do it. This is probably a very proper commercial business transaction but there is no provision in law to permit such advance obligation of the Federal Government, certainly not without competition. The exchange agreement was signed on April 25, 1922, and provided for the exchange of crude royalty oil from Elk Hills for stored fuel oil in Pearl Harbor, Hawaii.

The final step was made on December 11, 1922, when a secret agreement was made under the preferential treatment provision of the former agreement whereby Pan American was to exploit the remainder of Reserve Number 1 under certain restrictions.

In addition to the two major leases described above there were others of a minor nature, affecting small portions of both Reserves Number 1 and 2. The two which deprived the Navy of its reserves were the Mammoth and Pan American leases.

The details of events which followed are sufficiently well known to require little further exposition. The entire matter was thoroughly investigated by a Senate committee and special counsel was appointed by President Coolidge, in compliance with a Senate resolution, to prosecute all cases of fraud and to effect the return of the Petroleum Reserves to the Navy.

There were both criminal and civil court proceedings, as a result of the actions of the special counsel, in connection with the reserves. It was by means of the latter that the exploiters were removed from the reserves which then returned to Naval custody. Secretary Denby resigned in 1924 and his successor, Curtis D. Wilbur, immediately took steps to place all matters in connection with the reserves under his immediate supervision. Prior to this, Secretary Denby had designated Admiral Robison, Chief of the Bureau of Engineering, as the Department's representative in such matters.

It is interesting to note in reports of the Secretary of the Navy prior to Mr. Denby, that is those of Josephus Daniels, the petroleum reserves were accorded considerable space, largely recommendations as to the policy which should be pursued in connection with them. In Denby's reports they are not mentioned at all. Since 1924, each annual report of the Secretary of the Navy has included a detailed report of the status and condition of the reserves.

The litigation in connection with the reserve leases included two principal cases:

The Mammoth Oil Company lease to Teapot Dome was attacked by special counsel in Wyoming in 1925. On June 19, 1925, the United States District Court found in favor of the defendants, that is, that the leases had been legally executed and that the Mammoth Oil Company was entitled to exploit Teapot Dome. Government counsel appealed the decision and on September 28, 1926, won a reversal of the District Court decision in the United

States Circuit Court of Appeals which sat in St. Paul, Minnesota. Mammoth carried this second decision, which was against it, to the United States Supreme Court. This tribunal upheld the last decision in a decision rendered on October 10, 1927.

The Pan American lease to Elk Hills which was entered into in December, 1922, was attacked by special counsel in California. On May 25, 1925, the United States District Court in Los Angeles found in general in favor of the Government. Both parties appealed. The decision of the United States Circuit Court of Appeals in San Francisco was more sweepingly in favor of the Government than was that of the lower court. This latter decision was rendered in January, 1926, and was confirmed by the United States Supreme Court on February 28, 1927.

During the period in which the legality of the two major leases was being determined by the courts, the property in Elk Hills and in Teapot Dome which was under litigation, was administered by receivers appointed by the courts. In each case there were two receivers, one nominated by the lessor and one by the Navy Department. In Elk Hills the receivers were: Mr. J. Crampton Anderson, nominated by the Pan American Company, and Rear Admiral H. H. Rousseau (Civil Engineer Corps), U. S. Navy, nominated by the Government. In Teapot Dome they were Mr. Albert E. Watts for the Mammoth Oil Co. and, originally, Rear Admiral Joseph Strauss, U. S. Navy, for the Government. Admiral Strauss served from March 13, 1924, until August, 1924, when he was relieved by Commander (now Rear Admiral) H. A. Stuart, U. S. Navy. Admiral Rousseau and Commander Stuart served as receivers until the receiverships were terminated by the courts when the litigation was completed.

On March 17, 1927, the President issued an executive order, which revoked the 1921 one of President Harding, returning the administration and conservation of the Naval Petroleum Reserves from the Secretary of the Interior to the Secretary of the Navy. The formal transfer from one department to the other was effected on August 1, 1927. The receivership on Reserve Number 1 was terminated on August 26, 1927, and that on Reserve Number 3 on January 7, 1928.

The Naval Petroleum Reserve Office was established in the Navy Department by Secretary Wilbur in 1924. This office continued under the Secretary's personal supervision until October 15, 1927, when Rear Admiral Rousseau was appointed Director of Naval Petroleum Reserves and Oil-shale Reserves and Captain Stuart was appointed Assistant Director. That office has continued under the head of a Director since that date. Admiral Rousseau died suddenly on July 24, 1930, and was succeeded by his assistant, Commander Thomas Moran, U. S. Navy, as acting director. Rear Admiral F. T. Chambers (Civil Engineer Corps), U. S. Navy, took over the Directorship on October 6, 1930. Admiral Chambers continued in this office until his death in November, 1932. Commander Moran again acted until he was relieved by Captain H. A. Stuart, U. S. Navy, in June, 1933. Captain Stuart is the present incumbent.

When The Petroleum Reserve Office was established, Offices of Inspector of Naval Petroleum Reserves were also set up, one in Los Angeles, California, for Reserves Number 1 and 2; and one in Casper, Wyoming, for Reserve Number 3 and the oil-shale reserves. The first inspectors were Lt. Com. Irwin F. Landis, U. S. Navy (retired), in Los Angeles and Lieutenant Commander W. H. Osgood, U. S. Navy, in Casper. Landis had been employed in connection with the reserves before the war as mentioned in a previous part of this history. He continued as inspector until the Economy Act of 1933 forbade the employment on active duty of retired officers. His successors have been:

Rear Admiral W. C. Cole, U. S. Navy, 1932
Captain H. A. Stuart, U. S. Navy, 1932 to 1933
Commander R. M. Comfort, U. S. Navy, 1933 to 1936
Commander W. G. Greenman, U. S. Navy, 1936 to ?

The Officers who held the position of Inspector at Casper were:

Lt. Comdr. W. H. Osgood, U. S. Navy, 1924 to 1928
Lt. Comdr. C. A. Trexel (CEC), U. S. Navy, 1928 to 1930
Lt. Comdr. A. A. Baker (CEC), U. S. Navy, 1930 to ?

When the Secretary of the Navy received the custody of the reserves back from the receivers and turned them over to the

Director of Naval Petroleum Reserves for administration and conservation, the following net results accrued to the Government from the series of transactions:

FROM TEAPOT DOME:

1. The most accurate and reliable information about Reserve Number 3, including an estimate of the total original recoverable oil content of the reserve of 20,000,000 barrels. This was obtained at a cost of 3,550,228 barrels of oil depletion.
2. Fuel oil tanks at Portsmouth, N. H., with a capacity of 300,000 barrels.
3. Clear title to all of the lands in Reserve Number 3.
4. Improvements on the reserve to the value of about $5,000,000.
5. $5,909,218.59 in cash and bonds as the value of the oil removed from the reserves plus interest at 7 per cent.

FROM ELK HILLS:

1. Same as 1 above with an estimate of original removable oil of about 600,000,000 barrels. The cost was a depletion in the reserve of about 20,000,000 barrels.
2. Fuel oil tanks at Pearl Harbor, Hawaii, with a capacity of 4,200,000 barrels.
3. 1,500,000 barrels of fuel oil in storage in Pearl Harbor.
4. Improvements on the reserve to the value of $1,407,151.99.
5. $24,237,341.05 in cash and bonds.

The total net gain to the Navy from the entire affair appears to have been quite appreciable. In April, 1921, when the reserves were transferred to the care of the Interior Department, the Navy had, disregarding Reserve Number 2, two reserves about which very little detailed information was on hand but which had an estimated recoverable content of between 250,000,000 and 300,000,000 barrels. There were claims to many parts of these reserves which had not been quieted, and the reserves were practically untouched as far as development was concerned.

The returned reserves were somewhat damaged, it is true, but the Navy now had full information as to the geologic and operating conditions which will be of great value at any time that it may

become necessary to develop the reserves. The Navy now had a very accurate estimate of the original recoverable oil which was at least twice as great as had been thought before. Of this entire amount about 3 per cent had been removed. On the reserves were several millions of dollars worth of improvements, some of immediate value and many others with future value when the reserves should be operated.

The Navy also gained fuel oil storage at strategic locations, storage which had been sorely needed for some time and even not yet adequate at this time. The gain to the Navy in the total of fuel oil storage capacity was more than twice as much as had ever been legally built using regularly appropriated funds.

Most important of all of the gains which the Navy and the Government made as a result of the leases was the fact that all doubts as to title were now removed and an aroused public practically assures that no further attempts to wrest its reserves of oil from the Navy will be made for many years to come, if ever again.

In 1924, in addition to other steps taken by Secretary Wilbur in clearing up the reserve situation, he had recommended to the President that a commission to study the reserves and to outline a policy for their future handling be appointed. This was done and the Presidential Oil Commission consisting of Dr. George Otis Smith, Director of the United States Geological Survey, Rear Admiral Hilary P. Jones, United States Navy, and Mr. R. D. Bush of the California Bureau of Mines was formed. This commission rendered a report whose general and principal recommendation was that Reserves 1 and 3 should be shut down completely. This was done as soon as the receiverships were terminated and the Navy was free to take control. In the event that this shutting down should result in a loss of Navy oil from the reserves through offsetting border wells, agreements were made with owners of such offset wells to pay an increased royalty to the Government in the event that the shutting down of Navy wells resulted in an increased production from their wells. The gain in production of these offset wells was inappreciable. Studies since the reserves were shut in indicate pretty conclusively that there has been very little drainage of reserve oil except probably during the flush production stage.

On January 22, 1927, the Presidential Oil Commission submitted a recommendation for certain legislation. The matter was presented to the House Naval Affairs Committee on March 1, 1928, and a bill " for the conservation, care custody, protection, and operation of the naval petroleum reserves " was introduced on December 21, 1929. The Navy Department urged the passage of this bill consistently until it was finally enacted into law on June 30, 1938. Ten years is a long time to get essential legislation passed but the Naval Petroleum Reserves had become a highly publicized subject which was very difficult to touch upon. The feature of this legislation which was most highly desirable was the provision that the Secretary of the Navy could exchange land within Reserve Number 1 which was in private ownership for other land in Reserve Number 1, or for royalty production from any petroleum reserve. This provision will make possible the consolidation of this greatest of all reserves in such a way that its future availability for emergency use will be certain. In the event that agreements to make the necessary exchanges cannot be reached, the Secretary is authorized to initiate condemnation proceedings.

The oil situation was very much to the fore in 1924 and the fact that the general subject was much broader than the question of the Navy's supply of fuel oil was recognized. The Nation had a natural resource upon which it was becoming more and more dependent. It was a depletable resource and the cry had been raised several times that it was being wasted. The President, in December, 1924, appointed the Federal Oil Conservation Board to consider the problems of the country's petroleum industry. This board consisted of the Secretary of the Interior as chairman and the Secretaries of War, Navy, and Commerce as additional members.

The Federal Oil Conservation Board was entirely without power to issue any edicts, to make any rulings, or to punish any wasters of petroleum. Yet, acting entirely through cooperation with the recognized leaders in the petroleum industry, it effected a tremendous good which will be reflected in the supply and price of petroleum products in the future. The benefits of the work of this board will accrue to the Nation at large. The Navy will receive

its share in the more certain availability of petroleum and its products and a probably smoother shift from petroleum to substitutes when the changeover becomes necessary. Due to the efforts of the board, aided and abetted by the interest of the people of the United States, and brought to fruition by the intelligent cooperation of the petroleum industry, it will be much longer before the Navy will be forced to depend on its reserve oil production than it would have been had the Board never been formed.

Substitutes for petroleum have been mentioned. The Federal Oil Conservation Board covered this subject quite thoroughly. Substitutes are available when needed and when economic conditions will permit or demand their development. The Navy's principal direct interest in substitutes has been concerned with oil-shale. This mineral rock is quite abundant and large areas containing it have been set aside as a secondary oil reserve for the Navy.

Naval Oil Shale Reserve Number 1 consists of 36,550 acres located in Colorado. It was withdrawn from entry and established as a Naval reserve on December 6, 1916. Reserve Number 3 surrounds Number 1 on three sides but itself contains no oil shale. It was withdrawn and set up as a Naval reserve on September 27, 1924, for the purpose of providing an adequate area for plant operations if and when it ever became necessary to mine and retort the shale. Reserve Number 2 is located in Utah and consists of 91,464 acres, all but 4880 acres of which was withdrawn and established as a Naval reserve at the same time that Number 1 was. The 4880 acres was added by Executive Order dated November 22, 1924.

There have been many people under the impression that oil shale contains already formed hydrocarbon oils similar to petroleum which can be removed from the rock by distillation. If this were the case it would be a relatively simple technical matter to obtain the oil. The fact is that there is no oil as such in the rock. All of the necessary chemical elements, that is: hydrogen and carbon are in the shale and they are in such form that it is possible to convert them into an oil which resembles petroleum in many particulars and to remove the oil thus formed from the rock.

The oil shale reserves were set aside at a time when powerful interests were hoping to get control of the petroleum reserves,

trusting that the Navy and the public would consider shale reserves as adequate for national defense purposes. It is probable that a great deal of misinformation was circulated at that time in regard to shale oil for propaganda purposes. At the same time and for some years afterward, there were a number of companies formed for the avowed purpose of producing shale oil. By far the majority of these were merely promotional schemes. Nineteen hundred and sixteen was a time at which real fears were entertained in the United States that the nation's petroleum would soon be exhausted and that substitutes would have to be produced unless the country was to become dependent on importations of oil to meet its needs.

Because of the general interest in the subject and the possibility that oil shale might soon become vitally important to the country, the United States Bureau of Mines began active work on the subject. Production of shale oil all over the world was studied and a great deal of laboratory work was done on different domestic shales and also on imported samples. Equipment which had been designed and patented in this country had had no practical test. There was no real commercial development any place in the world except in Scotland. Other shale oil developments which had been worked at times past had been forced out by the competition of cheap petroleum.

In 1924, the Bureau of Mines issued Bulletin 210 entitled "Oil Shale—An Historical, Technical, and Economic Study." This publication brought the literature on the subject up to date and indicated the course which further development in the United States should follow. One of the recommendations of the President's Naval fuel oil committee was that the Bureau of Mines makes a study of oil shale to determine its feasibility as a source of fuel oil. In November, 1924, representatives of the Bureau of Mines surveyed both of the Naval Oil Shale Reserves to locate a site for an experimental plant to produce shale oil. The site finally decided upon was located near Rulison, Colorado, with the proposed mine on Reserve Number 1 and the production plant on Reserve Number 3.

Ninety thousand dollars was appropriated in 1924 for the Bureau of Mines to go ahead with the experiment. In selecting the equipment to be used, the Bureau of Mines took advantage of

all of the information which it had obtained in the past and decided upon the installation of two different retorts. The first one selected is known as the Pumpherston Retort. It had been in successful operation in Scotland for many years and was considered the best by many in the Scottish industry. The special parts for this retort were purchased in England and the erection was supervised by a construction expert of the contractor who supplied the parts.

The Pumpherston retort utilizes external heat and for that reason it was desired that the second retort selected be one which used the direct heating process wherein the heat required for the pyrogenesis of the oil is supplied by the combustion of gas and carbon in the spent shale within the retort. Two small retorts of this type had been previously operated in this country. One, known as the "Dundas-Howes," or more commonly the "N-T-U," was selected and purchased from the N-T-U Company who erected the retort at the plant site.

The mine was put in operation and shale was ready from it when, on September 11, 1926, the tramway which was constructed to carry the shale from the mine to the plant was placed in operation. The Pumpherston retort was completed and began its first run in the morning of September 17, 1926. The first oil was produced during that night. The N-T-U retort began its first run on January 17, 1927.

The plant operated, producing oil until July 1, 1927. On that date the money available ran out and Congress had not appropriated for continuance of the plant. In January, 1928, $45,000 was appropriated to put the plant into condition for operating again and $75,000 was appropriated for operations during the fiscal year 1929. Operations were resumed in June, 1928, and continued until the Spring of 1929 when they were discontinued and the plant dismantled, partly due to lack of funds but also because the information which the Bureau of Mines had started out to get had been obtained.

The oil which was produced by the Ruliston plant was distributed to various laboratories but the greater part of it was sent to an experimental refinery which was set up cooperatively between the Bureau of Mines and the State of Colorado at Boulder,

Colorado; and to the Fuel Oil Test Plant (Naval Boiler Laboratory) at the Philadelphia Navy Yard. The purpose of the former was to determine the commercial values and possibilities of the product of the two retorts; and of the latter to study the value of the product as a Naval fuel oil in substitution for the petroleum fuel oils which had been the only ones previously used.

Only crude shale oil was shipped to Philadelphia. No burning or other tests were made by the Naval Boiler Laboratory on any residua from the refining of the shale oil.

The general results of the laboratory tests was that shale oil was a substitute for petroleum but was inferior for the production of specification petroleum products. The indications were, however, that when shale oil was economically needed, developments would very probably take place which would make the products entirely satisfactory.

On June 6, 1927, two tank cars of shale oil, one from each of the two retorts, were received in the Philadelphia Navy Yard from the Ruliston plant. These were the first and, as far as the author is aware, the only carload shipments of shale oil ever made in the United States. As previously stated, this oil was in the "as produced" state, not having been refined or treated in any way. This, in itself, marks the oil as exceptional as a fuel oil because in only a very few early instances had any burning tests under Naval boilers been made on any "crude" petroleum. In this respect, shale oil does not differ from petroleum. It can be refined to produce the products which demand the better market such as: gasoline, kerosene, lubricants, and light heating and Diesel oils. In any commercial operation it is very probable that any product which will be placed on the market as a bunker fuel oil will be a residual product from a refining process which removes the greatest value from the crude shale oil and hence will be quite different from the crude oil which was tested. Practically, this may affect only quantity available since the Navy is rapidly approaching the point where the usability of any possible fuel oil can be determined analytically without recourse to a boiler burning test. Furthermore, if the Navy is ever reduced to the production of oil from shale mined from its own reserves as a supply of fuel oil, there can be

little doubt that the urgency of the situation will permit of the use of crude shale oil as produced regardless of its commercial value.

The Naval Boiler Laboratory was directed to test the oil for the determination of the following points:

(*a*) The characteristics of the two types of crude shale oil.

(*b*) Their behavior in storage and in handling.

(*c*) Possibility of blending crude shale oil with fuel oil for Naval use.

(*d*) Type of mechanical pressure atomizer burner best suited for use with shale oils.

(*e*) The suitability of crude shale oil for Naval use as a substitute for petroleum fuel oil.

The laboratory's test report was submitted on September 2, 1932, and included a very thorough abstract which is quoted below in full:

"The test was conducted over a period of nearly five years to determine the characteristics, analyses, blending with bunker 'C' petroleum fuel oil as well as comparative evaporative efficiencies versus Bunker 'A' petroleum fuel oil when using five types of fuel oil burners.

Two types of crude shale oil produced from oil shale mined at Rulison, Colorado, were tested. Though identical in origin each differed in characteristics due to method of distillation.

(a): Shale Oil F.O.T.P. No. S-874 distilled in Pumpherston (Scotch) retort.

(b): Shale Oil F.O.T.P. No. S-875 distilled in 'N-T-U' (Dundas-Howes intermittent down-draft type) retort.

Chemical, physical and thermal characteristics in general are similar to those of petroleum fuel oils.

Deterioration during normal periods of storage would not be excessive.

Tanks, lines and strainers require heating during normal winter operation to prevent wax deposition.

Low viscosity of the Pumpherston retort crude will cause excessive slip in certain types of pumps necessitating precooling for normal summer operation.

Shale oil can be successfully blended with Bunker 'C' petroleum fuel oils, thus little difficulty should be experienced with normal shale-petroleum fuel oil mixture with existing fuel oil burners. They burn faster, with shorter flames and require less air velocity than petroleum fuel oils.

Shale oil is considered satisfactory for Naval use ashore and afloat as a substitute for petroleum fuel oil.

Shale oils are recommended for Naval use as petroleum fuel oil substitutes in event of shortage of the latter. To establish further potential substitutes, possibilities of other liquid non-petroleum fuel oils, such as water-gas tar, should be investigated."

The results of these tests can be paraphrased and converted into terms fitting the broad problem of the use of fuel oil as follows. Considering the two oils together, they differed from the fuel oils which the Navy had been getting only in that they were more fluid, contained large quantities of paraffine wax not usually found in fuel oils, and contained much greater percentages of sulphur and nitrogen which in no apparent way interfered with the full utilization of the oils as Naval fuel oils. Only the first two mentioned differences had any effect as far as Naval usage was concerned. The greater fluidity of the oil permitted one of them (Pumpherston) to be efficiently atomized without any preheating whatever and the other (N-T-U) with very little preheating. Except for gasification this is a considerable advantage particularly in view of what has since been discovered about the ill effects which are possibly obtained in fuel oil heaters. Slight modifications in burning equipment should be possible, to remove any adverse effects from gasification at high burning rates. Slight redesign of pumps should eliminate any difficulty such as that noted in the excessive slip and overheating in pumps.

The presence of excessive quantities of paraffine wax does present greater difficulties than the report of the Fuel Oil Test Plant indicates on its face. Apparently difficulties were experienced whenever the temperature of the storage tanks or of any part of the piping system fell below a point in the neighborhood of 40 degrees Fahrenheit. It is very rarely that this condition would be expected to obtain in shipboard installations and yet it is possible

in a fleet which must of necessity be an all-season, all-weather, all-latitudes organization. Any difficulties from this cause could quite simply be guarded against by a properly designed installation. However, the remarkable results which are obtainable in refining of petroleum indicate that the simplest all-around solution would be one which would either remove the wax, lower its point of crystallization, or neutralize its effect before the fuel oil is delivered for use. This should be very easy and cheap.

Another characteristic of crude shale oil not mentioned in the abstract, and one which also applies to *crude* petroleum is that the flash point of much of it is less than the specification requirement, in effect since 1914, of 150 degrees Fahrenheit. The Laboratory reported no difficulties, either realized or expected on this score. It would be conceivable that the specification requirement might be lowered to admit the lower flash point of crude shale oils but it would be almost as simple, and offer more assurance to operators, to adhere to the present limitation and subject the shale oil to a very light topping process which would raise the flash point.

The above discussion leads to the conclusion that: *with a very slight and easy modification of burning and pumping equipment in Naval vessels and with a very simple and inexpensive refining process to raise the flash point and to lower the pour point, all preheaters and storage tank heating coils in Naval vessels could be eliminated and the fleet could be supplied with a "modified" crude shale oil which would be a superior Naval fuel oil for all climates and rates of steaming.*

The preceding conclusion is based entirely on engineering grounds. It completely disregards economics, and this side of the proposition has been overlooked by many people who appear to believe that by a mere shift of valves, the Navy can start using shale oil when the production of oil wells begins a serious decline. The economic difficulty can be reduced initially to two considerations. One is the comparatively low cost of petroleum products which is due almost entirely to the PRESENT apparently unlimited quantities in which it can be produced. The other is the estimated enormous capitalization cost of bringing shale oil into production in quantities sufficient to constitute a market factor.

In recent years, the bulk of all petroleum produced has sold at the well at a price which rarely exceeds one dollar per barrel by more than a few cents. During the period when the receivers were operating the reserve leases which were in litigation, their total cost of operation was about one-sixth of the value of the oil produced. Thus for an operator who was producing oil from land which he owned in fee and who had no other connections whose losses had to be covered, it is conservative to state that more than 75 cents per barrel above the cost of actual production of $1 a barrel oil is available to cover the cost of exploration, leasing or purchase, and development. This does not necessarily indicate tremendous profits but it has resulted in many actual cases of profits so large as to have been fantastic. There is no such possibility with shale oil. When and if it comes as a commercial venture it will require the very best management to pay the most conservative of profits.

It has been estimated that the capital cost of producing crude shale oil would require an investment for plant of about $3000 per barrel per day of capacity. This estimate was made years ago when costs were much lower than they are today. Even this figure, assuming 6000 days amortization, and 4 per cent interest on investment per year would impose a capital charge of more than 80 cents on each barrel of oil produced. Estimates of operating costs have been made freely only by promoters who have in some cases claimed that all costs would permit free competition with petroleum. This is not true today and very probably will never be true until the price of petroleum at the site of the oil shale exceeds at least two 1926 dollars. This figure is given grudgingly for purposes of discussion but is believed to be a low minimum.

The economics as presented above remove shale oil from present consideration as a substitute for petroleum fuel oil for Naval use on normal grounds using an expenditure dollar as the basis of comparison. However, there is one other basis which should be taken under advisement and seriously considered. There is no escaping the fact that some day, near or distant, the production of petroleum in the United States will begin a decline from which it will never recover. Beginning at that time the use of petroleum products measured volumetrically must also decline or we must

begin an increasing dependence on imported petroleum, or the supply of petroleum products must be maintained by the appearance of satisfactory substitutes on the market. Whatever develops it can hardly fail to present a problem to the Federal Government. Preparation could be begun at any time to meet the situation by continuing the work by the Bureau of Mines on the Naval Oil Shale Reserves and expanding this work until shale oil suitable for use as Naval fuel oil is being produced in sufficient quantities to take care of all of the Navy's operating needs.

The above proposition probably appears to be fantastic. It would involve a considerable expenditure which would not be self-liquidating until such time as the supply and demand for petroleum forced the price of fuel oil well beyond anything immediately in sight. What could be considered the loss of capital and the operating loss of the scheme would actually be a dollar loss only and could easily pay dividends not measurable directly by our monetary system, greatly exceeding the dollar cost. These dividends would be, to the Navy: eventual freedom from all difficulties which the present system imposes in changing characteristics of fuel oil and problems of supply. By having the entire production and distribution under Government control, the further difficulties which the Government contract system imposes would be removed. Since the Navy's current demand for fuel oil is considerably less than 1 per cent of the total volume of petroleum products marketed and much less than that percentage in value, the competitive factor with private business would be immaterial.

Both the public at large and the oil companies would gain by having a real start made in the production of shale oil. A large portion of the loss could honestly be charged to a subsidy in establishing an infant industry which would be in a position to expand in private hands when shale oil was required to keep the country's machinery lubricated and its automobiles and airplanes moving. Perhaps the present is not the time for the venture but careful timing will be required to have the petroleum substitute development at the proper stage when it is required. Beginning it by the Government through the medium of the Navy's fuel oil needs and the Navy's Oil Shale Reserves appears to be a way of doing

it with the elimination of much of the speculation and uncertainty which attends new enterprises.

In its report on petroleum substitutes, the Federal Oil Conservation Board recommended that a Naval Coal Reserve be set aside because, in its opinion, coal would be utilized before oil shale as a source of petroleum-like oils. This reserve was not set aside so the Navy is not now in a position to consider the use of coal for its own production of fuel oil. Whether industry will turn to coal ahead of oil shale for petroleum substitutes remains to be seen.

On February 27, 1923, the President set aside a fourth Naval Petroleum Reserve. Reserve Number 4 is located in Alaska and has an area in excess of 22 million acres. It is in the northern part of Alaska beyond Point Barrow and may or may not, at some time, produce petroleum. Oil seepages are known on this reserve and it is assumed that the area is petroliferous. From 1924 into 1927, the United States Geologic Survey sent exploring parties into this reserve. Most of the work was financed from Naval appropriations. The Survey issued Bulletin 815 in September, 1930, describing the conditions existing in the reserve. There is a possibility that, even though great quantities of oil may exist in this reserve, the cost and extreme difficulties which will attend its production will make Reserve Number 4 of less future value than the oil shale reserves.

It is probable that the source of the Navy's fuel oil at any time will be governed solely by dollar economics. That is, the Navy will be required to operate on the fuel which can be most cheaply purchased in the commercial market. No workable plan for the future can be made and established without increased appropriations. Legislation exists which authorizes the Secretary of the Navy to operate any of the reserves. There is an appropriation which permits the use of any other money appropriated to the Navy not to exceed ten million dollars for the development and production of the petroleum reserves if necessary to protect them. Money is not available for developing the oil shale reserves.

This probable dependence on the sources of petroleum which are under commercial control makes the Navy vitally interested in all questions of petroleum exploration, discovery, production, and demand and any other questions which may affect either the

volume or the characteristics of the product which is marketed as fuel oil. It will be because of studies of these matters that the Navy will have to make the decision to produce its own oil. As the reserves are brought into use, the next question will be whether the Navy can hold their entire product for its own use or will have to use it to hold up the decline of the general supply. In the latter event, not much time will be permitted for undertaking the development of oil shale.

Reserve Number 1 could be produced to supply the Navy's needs for all petroleum products for a matter of about thirty years and during that period should be able to handle the Naval demand for a year or two of wartime operations. If the Navy was forced to throw some of its oil on the commercial market, it is possible that Reserve Number 1 would be of little value after something like ten years of production.

Reserve Number 2 is at present of questionable value. The very great possibility that there are large quantities of untouched oil remaining in strata below those now producing may mean that the Navy owns a very valuable Reserve in Number 2. Much will depend on whether or not it will be possible to refuse to lease the lower strata and in addition on what arrangements can be made with the owners of the present land within the reserve to withhold their oil in the lower strata from production. It is probable that the best solution will be to permit continuation, under the most effective engineering conditions, of the development and production of Reserve Number 2 until the greatest depths attainable have been plumbed and the field depleted. The field can be controlled to prevent wasteful operation but if demand for petroleum requires the oil from Reserve Number 2, it is questionable whether the Navy, possessing Reserve Number 1, could properly hold in this oil.

Reserve Number 3 cannot be very seriously considered as a source of future supply for the Navy. The field is located a long way from sea water and does not contain enough oil to make it worth while to make the necessary provisions for producing this oil and transporting it to locations where the Navy can use it. As a matter of sound general economics it would probably be advisable to permit the oil from Teapot Dome to be used when nec-

essary to augment the supply from the Salt Creek field in order that full utilization may be made of the facilities which have been developed for handling that oil.

Reserve Number 4 may never be used, no matter how pressing the need for oil may become. In the first place it may contain no oil, and secondly, the practical difficulties of producing and transporting petroleum from a location which is so isolated by extreme winter weather, may place any oil in this reserve in a position of lesser availability than the oil from shale which is located within the United States proper.

The shale reserves could take care of the Navy's needs for an indefinite period after they are brought into production and since there is a great deal more land which is just as valuable for the production of shale oil, there should be no commercial interference with the use of its own oil by the Navy.

The best return to the Navy and to the United States from its petroleum and oil shale reserves will depend almost entirely on the question of timing. The most important decision which will have to be made in each case will be when to start development and production. Once that decision is made, circumstances and common sense will determine the rest.

SOME FACTS ABOUT CENTRIFUGAL AND PROPELLER PUMPS.

By Lieutenant Commander Rupert M. Zimmerli, U. S. Navy, Member.*

Introduction.

Twenty years ago nearly all of the water pumping auxiliaries in Naval vessels were of the reciprocating type. Today the only reciprocating pumps retained are for the emergency boiler feed and for the fire and bilge pump service. With these two exceptions water pumping is being accomplished by the use of centrifugal and propeller pumps. The author presents a short discussion of the latter types without going into the more intricate design details.

Centrifugal and Propeller Pumps in General.

Centrifugal pumps depend for their operation upon the high velocities imparted to the fluid pumped being converted into static pressure. The propeller pump does not depend upon the action of centrifugal forces, but pushes the water in an axial direction parallel to the line of the shaft. This latter type is frequently termed "axial flow" pump and is used primarily for large capacities and relatively low heads, such as for condenser circulating services.

Both of these types combine high operating speeds with large capacities as compared with reciprocating pumps. As with other types of machinery, high rotative speeds permit small size with relatively high output. Reduced weight and space requirements naturally follow. From this we see that the centrifugal pump or propeller pump answers the needs of the Naval service as it fulfills the basic requirements, i. e., high output, small space, and low weight. But these characteristics are worthless without maximum

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reliability, simplicity and accessibility for maintenance, and good operating economy. Reliability of the present centrifugal and propeller pumps materially surpasses that of the old style reciprocating pumps. Operating economy is a question of how many pounds of steam or how many pounds of oil are required to produce a horsepower output. Inasmuch as most of the essential auxiliaries are steam turbine driven and as the efficiency of the turbine depends almost directly on high turbine blade speeds, it follows that operating a centrifugal pump at high rotative speeds is essential for good steam economy. The efficiency of the pump itself is fairly good, having been increased from 20 to 35 per cent in small capacity pumps over those built 10 to 15 years ago, to as high as 65 to 85 per cent in medium to large capacity units of present design. Some of the factors which influenced the increased efficiency of these pumps were:

(*a*) Smoothing out abrupt changes in direction of flow of water within the pump.

(*b*) Fairing up all surfaces in contact with water so as to avoid any abrupt changes in direction or velocity within the pump which results in excessive shock losses.

(*c*) Providing liberal areas of all passages and ports within the pump casing and in the impeller so as to reduce internal friction losses to a minimum.

(*d*) Increasing the length of the passages in the impellers of pumps to give the required characteristic curves necessary for certain particular operating conditions. In some cases increase in rotative speeds was possible, while in other cases reduction in blade tip speeds for small capacity pumps resulted in considerable reduction in so-called disc friction losses.

(*e*) Improvements in types of impeller and casing wearing rings and sealing devices which resulted in reduction of internal leakage within the pump.

The basic design requirements previously mentioned, together with the keen competitive situation existing in the centrifugal pump field have resulted in high speed pumps of small physical dimensions but fairly large capacity and, when needed in some cases, of high discharge pressure. This has produced some de-

signs of pumps which have very little margin of capacity over rated capacity in spite of any increase in speed above rated speed. The following laws hold for any centrifugal pump:

(*a*) Capacity varies directly as the speed.

(*b*) Discharge pressure varies as the square of the speed.

(*c*) Horsepower varies as the cube of speed. All of the above variations occur simultaneously as the speed changes. It should be noted particularly here that these laws hold true only within a practical operating limit of the pump, and that operation beyond this limit may produce erratic performance and result in reduced head (discharge pressure) and capacity.

Characteristic Curves of Centrifugal and Propeller Pumps.

The simplest way of illustrating the idiosyncrasies of the centrifugal and the propeller pump is by a discussion of their characteristic curves showing the effect upon their operation of speed, submergence and temperature of water handled.

Figure 1 shows constant speed characteristic curves of a 400-GPM pump designed for operation at a speed of 4500 RPM when handling 170 degrees F. water at a 4-foot submergence and designed to discharge against a pressure of 460 pounds gage. These curves indicate that at rated speed of 4500 RPM the pump will deliver about 450 GPM with stable operation, and that with speed increased to 4950 RPM the expected increased capacity and head is obtained only to a certain point (480 GPM) at 500 pounds pressure. Beyond this point the pump will be unstable so that the maximum capacity of this particular pump will be 480 GPM, regardless of the speed at which it is run, as long as submergence and water temperature are maintained constant. In this region we may say that the pump will be cavitating or running in cavitation. This means that the pump impeller vanes are pushing the water out to the volute casing and the discharge nozzle so fast that the losses in the suction of the pump, the basket or suction chamber of the casing and the eye of the impeller, are greater than the submergence head available and the pump pulls the water apart in the eye or in the vicinity of the eye of the impeller, creating a partial vacuum at that point. What is the solution of the trouble?

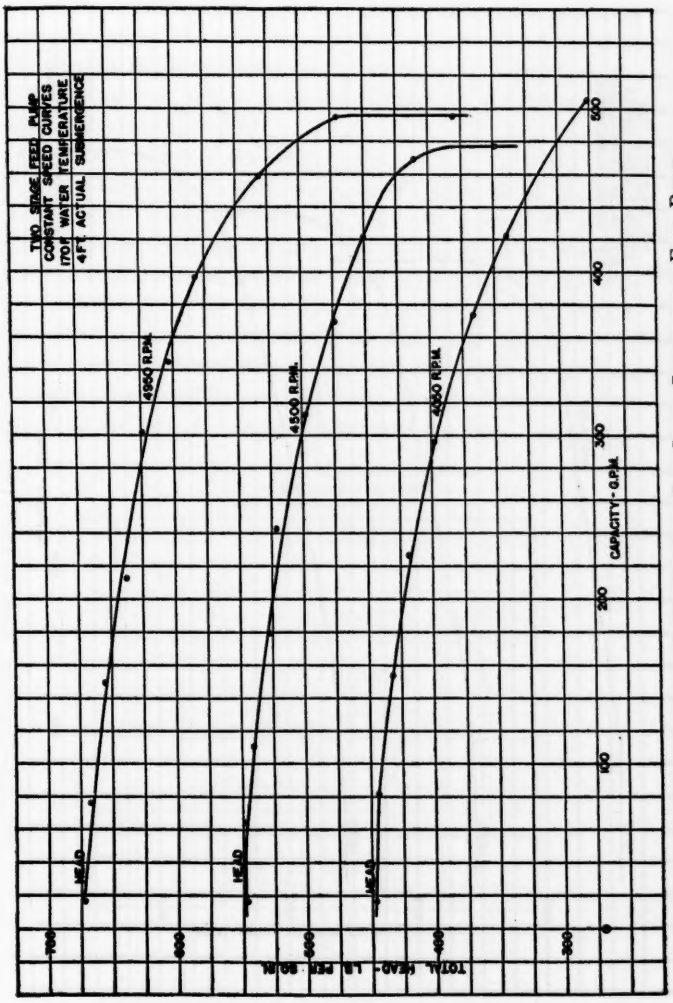

FIGURE 1—CONSTANT SPEED CURVES—TWO-STAGE CENTRIFUGAL FEED PUMP.

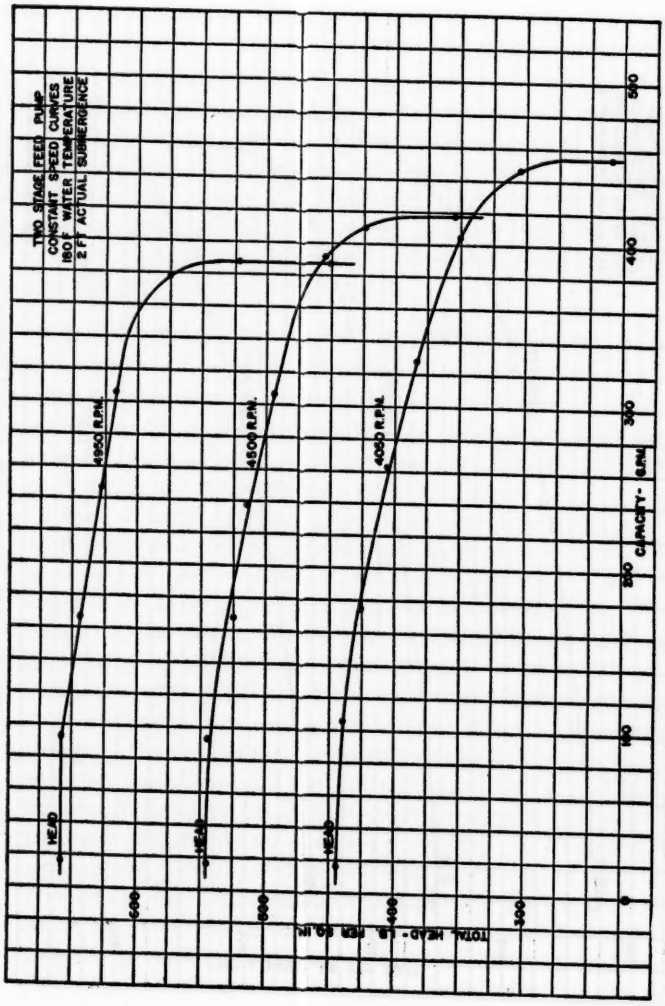

FIGURE 2—CONSTANT SPEED CURVES—TWO-STAGE CENTRIFUGAL FEED PUMP.

(1) Design the pump with less inlet losses, i. e., more liberal areas in the suction passages and in the impeller, particularly at the eye, or

(2) Reduce the temperature of the water, or

(3) Increase the pressure on the suction of the pump, i. e., increase the submergence, or

(4) A combination of the above.

Figure 2 shows the characteristics of the same pump operated at 4950 RPM, 2-foot submergence and 180 degrees F. water. In this case the maximum capacity has been reduced from 480 to 384 GPM at 500 pounds, also the pump is unstable, i. e., the discharge pressure drops off abruptly. This series of curves demonstrates that with each speed increase the maximum capacity of the pump is reduced rather than increased as we would expect from the laws of centrifugal pumps. Basically, the trouble here is *water too hot* and *submergence* (suction pressure) *too low.*

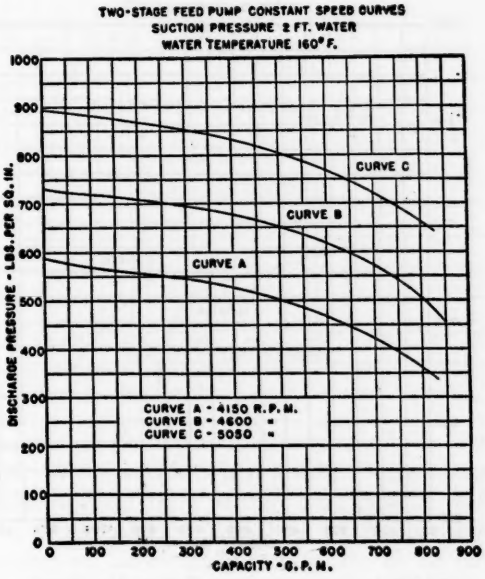

FIGURE 3—CONSTANT SPEED CURVES—TWO-STAGE CENTRIFUGAL PUMP.

To show what liberal design will do, the characteristic curves of another pump are illustrated in Figure 3. This shows the characteristics of a pump design for 600 GPM, 575 pounds discharge pressure, 4600 RPM, 2-foot submergence with 160 degrees F. water. The pump developed close to 700 GPM at this design condition. With speed increased to 5050 RPM, a capacity of 825 GPM was developed at 640 pounds discharge pressure with a probable maximum capacity of about 900 GPM indicated. However, the maximum capacity is problematic, for comparison of the slopes of 4150-RPM Curve A and 4600-RPM Curve B demonstrates that the B curve is just commencing to break at about 800 to 850 GPM; and it is questionable that this pump could have a maximum stable capacity of much more than 850 GPM (note divergence of Curves B and A from parallelism beyond the 700-GPM point).

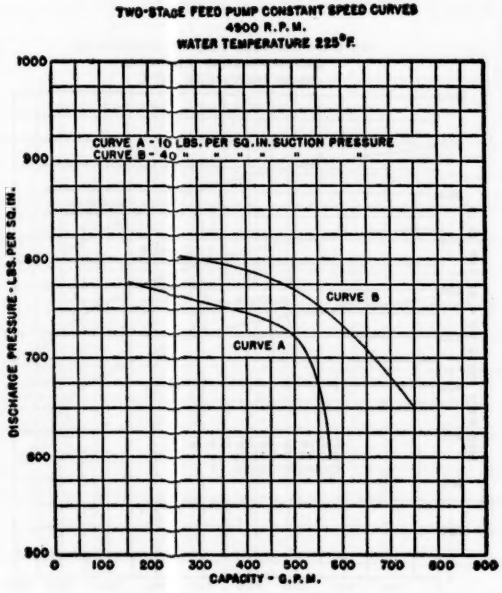

FIGURE 4—CONSTANT SPEED CURVE—TWO-STAGE CENTRIFUGAL FEED PUMP.

To illustrate more clearly the effect or submergence of suction pressure and temperature of water on the operating characteristics of a pump, Figure 4 shows the characteristics of the same pump when operated on 225 degrees F. water with suction pressure ranging from 10 pounds, as shown on Curve A, to 40 pounds, as shown on Curve B. At 4900 RPM the maximum capacity is about as before, i. e., 750 to 800 GPM with a 40-pound suction pressure, but with suction pressure reduced to 10 pounds at the same speed the maximum stable capacity is only 450 to 500 GPM. The above discussion applies particularly to boiler feed pumps, but the principle brought out applies equally well to all centrifugal pumps, except possibly condensate pumps.

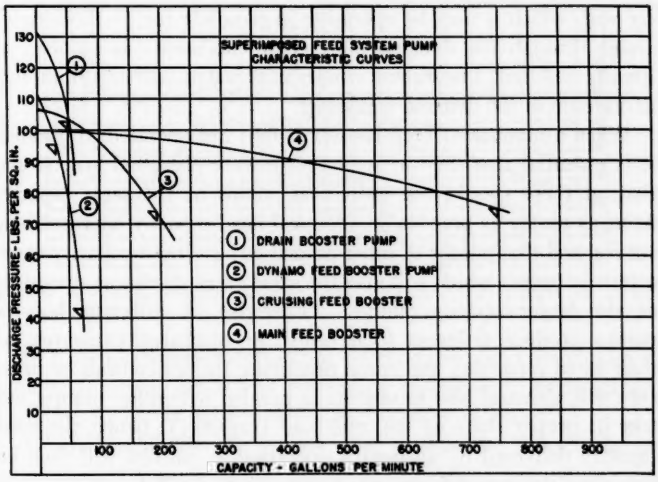

FIGURE 5—BOOSTER PUMP HEAD CAPACITY CURVES.

Feed booster pumps, Figure 5, usually operate at or near constant speed. With constant or near constant operating speed these pumps work back and forth on their head capacity characteristic curve, i. e., as the capacity demand on the pumps increases, the discharge pressure automatically reduces, and as the capacity demand reduces, the discharge pressure automatically increases. The

main, cruising, and dynamo feed booster pumps are designed to operate in parallel with each other and in series with the boiler feed pumps. In order that the feed booster pumps may operate satisfactorily in parallel without surging, the head capacity curves of these pumps must rise continuously from rated capacity to shut-off condition. For pumps having different rated capacities to operate in parallel as, for instance, main feed booster pumps and dynamo feed booster pumps, the shut-off head of the lower-capacity pump should be slightly in excess of the shut-off head of the larger-capacity pump. This is essential in order that the smaller-capacity pumps, which discharge through the dynamo air ejector condensers, may not be shut off the system by a higher discharge pressure developed by another feed booster pump of larger capacity with which it may be operating in parallel; for, if the flow of water through the dynamo air ejector condensers is stopped or greatly reduced, insufficient condensing water will result in loss of vacuum in the dynamo condenser.

When a number of feed booster pumps are operated in parallel, it is essential that they be operated at their designed speeds and that, at these speeds, the individual pumps of the same capacity have, as nearly as practicable, identical head-capacity characteristic curves, and that pumps of other capacities have head-capacity characteristic curves which will give the same rated discharge pressures as the larger-capacity pumps. Otherwise, if different booster pumps produce different total heads, or discharge pressures, at their rated capacities, the pump producing the lesser head or pressure will, although able to operate in parallel with other pumps, not take its proper share of the total load (capacity); hence, the total capacity delivered by all pumps operating in parallel will then be less than the sum of the rated capacities of all booster pumps operating.

In order to reduce the number of auxiliaries required for the main plant to a minimum, the feed booster pump and the condensate pump are frequently combined in one unit driven by a common driving turbine or motor. Booster pumps are usually designed for a capacity of anywhere from 10 to 25 per cent in excess of the feed pumps which they service, and for discharge pressure of from 75 to 85 p. s. i., with a head-capacity characteristic curve such that

by running a constant speed the pump will be able to develop a capacity of as much as 50 to 60 per cent in excess of rated capacity at a discharge pressure of about 40 pounds, which is ample to take care of friction loss in piping, air ejector condenser, feed heater, etc., and still leave a pressure of 15 to 35 pounds at the feed pump suction, a liberal margin of pressure above pressure equivalent to temperature, usually 225 degrees at feed pump suction. Thus any possibility of feed pump vapor binding or losing suction is avoided.

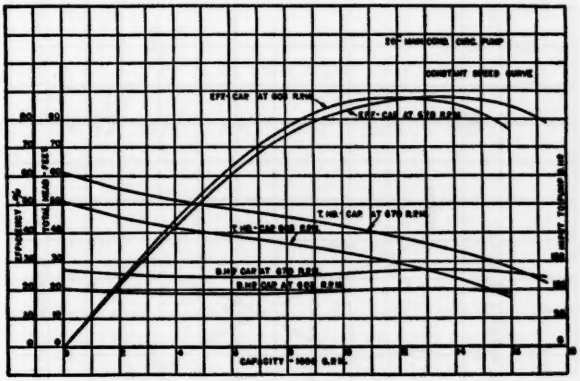

FIGURE 6—CHARACTERISTIC CURVES—MAIN CIRCULATING PROPELLER PUMP.

In a main condenser circulating water system the friction loss in condenser tubes, heads and circulating piping, etc., increases approximately as the square of the rate of flow of water.

Where circulating pumps are used exclusively for circulating water supply the pumps are usually of the centrifugal volute type, turbine driven, either direct or through a reduction gear. Reduction gear drive is usually preferred, as it enables both pump and turbine to be operated at their most efficient speeds. Where circulating pumps are installed to operate in series with scoops, single-stage propeller pumps are always installed, Figure 6.

The rising capacity horsepower characteristic of propeller-type pumps make them poorly suited for circulating water service when

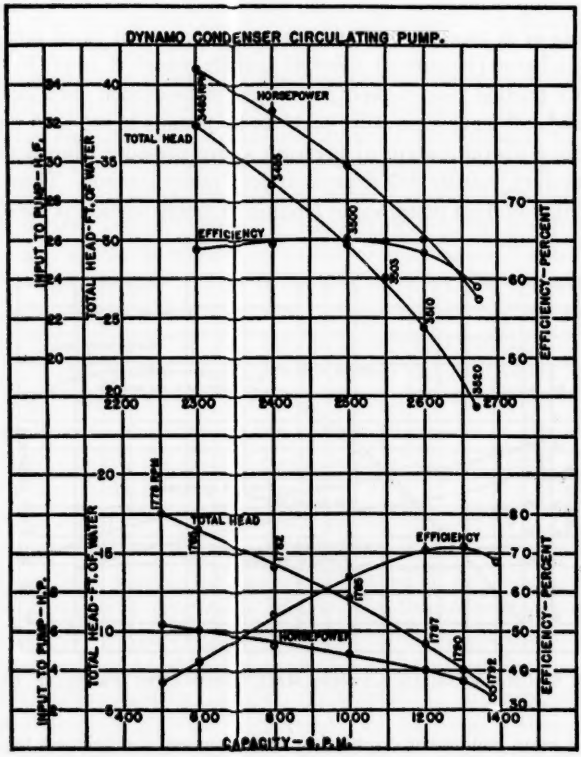

FIGURE 7—HEAD-CAPACITY CHARACTERISTIC CURVES. TWO-SPEED PROPELLER-TYPE PUMP FOR DYNAMO SERVICE.

driven by single-speed motors; for, if but a slight error is made in computing the total head against which the pumps must work, the motor may be heavily overloaded. Again, if marine growth clogs or restricts the flow of water through sea chest strainers, or if the condenser tubes are partially plugged with mud or sand, the result will be the same. This type of pump is ideally suited to steam turbine drive, as the infinite speed range obtainable with this type of drive makes possible any head and capacity within the limits of speed and horsepower of the turbine.

Miscellaneous Circulating and Supply Pumps.

The majority of pumps for duties such as flushing, dynamo circulating, etc., for capacities up to about 1000 gallons per minute at total heads or discharge pressures up to 50 pounds, are usually of the single inlet centrifugal volute type, close-coupled to the driving unit, usually an electric motor. In numerous cases, for example evaporator feed and evaporator brine pumps or distiller circulating and distiller condensate pumps, two pumps are driven by a common motor, the motor shaft being extended through both ends of the motor with a pump mounted on each end. For units above a capacity of about 1000 gallons per minute and for combined fire and flushing pumps the units are usually of the double inlet, single stage centrifugal volute type with conventional two-bearing construction, i. e., with a shaft bearing on each side of the pump casing. For combined fire and flushing service a conventional single stage, double inlet volute centrifugal pump of two-bearing construction is provided with dual drive; i. e., one end of the pump shaft is connected through a coupling to an electric motor, while the other end is coupled to a steam turbine. Flexible couplings are always used to couple the three units and, on later vessels, a free-wheeling clutch coupling is frequently used so that when the motor is driving the pump the turbine is stationary. Occasionally, where pump capacity is upward of 2500 gallons per minute with low total head, propeller-type pumps are used; however, while this type of pump offers advantages of reduced weight and space, it has the disadvantages of a steeply rising power input-capacity characteristic curve, which may result in overloading the driving motor, as discussed before.

Condensate and Brine Overboard Pumps.

Condensate and brine overboard pumps differ from pumps for other services principally in the design of the suction chamber and eye of the impeller. In both of these parts the passages and ports are made as large as possible in order to keep the velocity of water into the eye of the impeller down to a minimum. Great care must be taken with the layout of suction piping to see that the space directly above the eye of the impeller will vent itself back to the

condenser or evaporator shell. If this cannot be done, a suction chamber must be provided with a vent connection of liberal size piped with a continuous rise, without pockets or traps, to a point on the condenser or evaporator shell well above the highest water line and preferably to the point of lowest absolute pressure. If vapor is trapped or pocketed in the pump suction chamber or suction pipe, the flow of water to the pump will stop until the water level rises high enough to build up a pressure sufficient either to compress or condense the vapor and force it into the impeller.

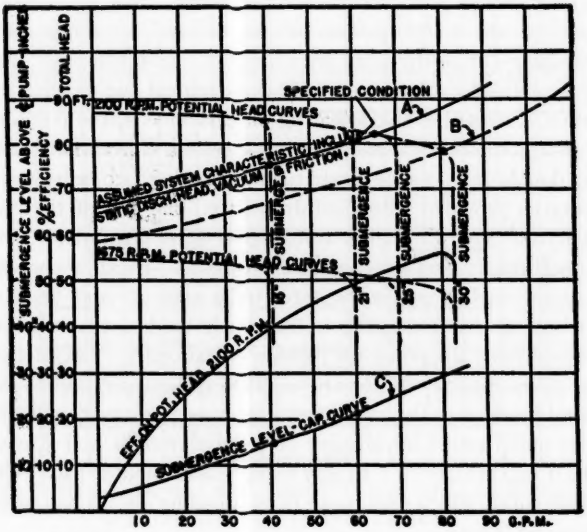

FIGURE 8—CONDENSATE PUMP CHARACTERISTIC CURVES.

Condensate pumps are usually run at very nearly constant speed. This is particularly true of pumps driven by alternating-current motors, such as cruising and dynamo condensate pumps and evaporator brine overboard pumps. Thus, it will be noted that while a pump may be designed for a given condition of capacity, total head and submergence, the quantity of water which the pump will handle varies widely, depending upon the amount of steam condensed in the condenser, or the amount of brining necessary to maintain

salinity in the evaporator. If a pump is to deliver water, the discharge pressure must be greater than the head against which the pump operates, i. e., the head capacity curve must be above the system head curve. The pump cannot operate to deliver water in the region where the system head curve is above the pump head capacity curve.

With a condensate pump operating at or near rated speed with fixed vacuum in the condenser shell, the primary factor governing the capacity at which the pump operates is the submergence. The manner of operation for the condensate pump is clearly shown in Figure 8 by the characteristic curves, which include a series of head capacity curves of a condensate pump for different submergence conditions, together with the system head curves against which the pump must operate. If we assume the system head curve to be as shown by Curve A, this particular pump, which is designed for 60 gallons per minute at 21 inches submergence, will, at rated speed of 2100 RPM, deliver 65 GPM, while the maximum capacity of the pump at 15 inches submergence is but 41 gallons per minute. If we assume a system head curve as shown by Curve B, then the pump will deliver a maximum capacity of 80 GPM at 30 inches submergence, and if the submergence is reduced, the capacity will be reduced, as in the system head Curve A. This reduction in capacity with change in submergence is more clearly shown by Curve C, which indicates how the capacity will vary with change in submergence, other operating conditions remaining fixed.

Conclusion.

The object of this paper is to acquaint the operating engineer, who has had neither the opportunity nor the time to make a special detailed study of centrifugal and propeller pumps, with a few of the inherent characteristics of the machinery under his care. It is hoped that a study of the presented characteristics will help the engineer intelligently to demand full service from this type of equipment with confidence that he will obtain the expected performance.

NOTES.

INDEX TO NOTES.

THE COMBUSTION GAS TURBINE: ITS HISTORY, DEVELOPMENT, AND PROSPECTS.
—Journal and Proceedings of the Institution of Mechanical Engineers, London, England, May, 1939.

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—Brown, Boveri Review, Baden, Switzerland, April, 1938.

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—Nickel Steel Topics, New York, N. Y., April, 1939.

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THE COMBUSTION GAS TURBINE: ITS HISTORY, DEVELOPMENT, AND PROSPECTS.

An article, "The Gas Turbine Challenges," by Lieutenant Commander Warren Noble, U. S. N. R., A-V(S), appeared in the May, 1939, issue of this JOURNAL. In it a preliminary survey was made of the application of gas turbines in the marine field. The following article was published the same month in the Journal and Proceedings of the Institution of Mechanical Engineers, London, England. The author is Adolph Meyer, E.D., Director, Messrs. Brown, Boveri and Company, Ltd., Baden, Switzerland. Dr. Meyer presents specific data and describes actual installations.

By "combustion gas turbine" is meant a turbine actuated by the steady flow of the products of a continuous combustion under pressure in a combustion chamber. Inventors appear to have been at work on the gas turbine since 1791, the original attractions of the proposal being its simplicity and the elimination of the reciprocating motion of the early steam engines. Simplicity remains the principal advantage of the gas turbine, though the first applications have been made possible by the needs of special chemical processes, such as the Houdry cracking process. The efficiency attainable under present conditions is 17-18 per cent, but this would be increased to 23 per cent if the gas inlet temperature could be raised from 1000 to 1300 degrees F. The proposed new fields of application of the gas turbine include locomotive and marine propulsion, blast furnace plants, and the power supply for wind tunnels.

Definition of the Combustion Turbine. The term "constant-pressure gas turbine," hitherto generally employed to denote a turbine actuated by the steady flow of the products of a continuous combustion under pressure in a combustion chamber, is inaccurate. The expression "constant-pressure turbine" was chosen to distinguish this machine from the constant-volume turbine, in which combustion takes place intermittently, by explosion in a closed chamber with a substantial rise in pressure. In the so-called "constant-pressure gas turbine" neither the combustion chamber pressure nor that before the turbine remains constant in service as, for instance, the steam pressure of a boiler; on the contrary, they depend on, and vary with, the load.

It is preferable, therefore, to call such a turbine a "continuous combustion gas turbine," or, briefly, "combustion turbine," in contradistinction to the explosion turbine.

History. The introduction of the gas turbine in the field of power generation is the realization of a long-cherished dream of engineers. Hardly any other kind of machine has received more attention from inventors, skilled and otherwise. A great number of suggestions has been made in regard to the method of operation of gas turbines, while patents covering design details and component parts run into thousands.

From the patent office records of different countries, inventors appear to have started tackling this problem at a very early date, for the first patent was granted in 1791 in England to one John Barber (Figure 1). Since then, the number of patents has steadily increased.

What may have induced the first gas turbine inventors to want to produce a substitute for the reciprocating steam engine, at that time the only form of heat engine in use? The question of efficiency was probably given but scant consideration, for the theory of heat was then little advanced. Two reasons can be suggested. First, the natural desire to replace the steam plant consisting of steam boilers, steam engine, and condensing plant, by more simple equipment, and, second, the wish to produce directly, by means of a turbine, the rotary movement (which, in the majority of cases, is necessary for the trans-

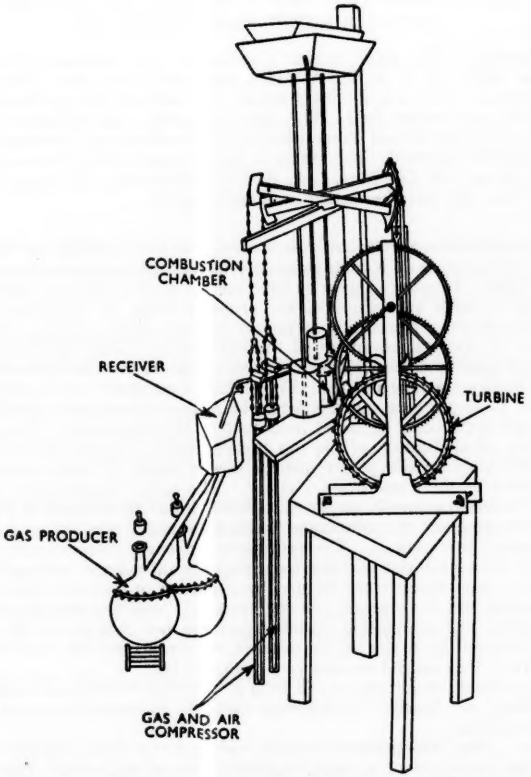


FIGURE 1.—DRAWING OF FIRST PATENT FOR A GAS TURBINE. (JOHN BARBER, 1791).

mission of the power), and to do away with the crank and connecting rod mechanism of the steam engine. That the desired simplification has been achieved can be seen from Figure 2, which shows the combustion turbine in its simplest form; B is the combustion chamber in which the gases are produced at a high temperature by the continuous combustion of some kind of fuel in an atmosphere of compressed air.

According to the fuel used, such turbines are referred to as gas, oil, natural gas, blast furnace gas, or pulverized-coal turbines. With all these fuels, the temperature of the combustion gases is high and must be reduced to a value compatible with the creep strength of the gas turbine blading. This can be done either by the addition of a large excess of air, or by the injection of water, or by a partial abstraction of heat by water-cooled surfaces, or other similar

means. Of these methods, cooling by injection of water must be ruled out of consideration, because of the loss associated with the latent heat of evaporation. The plant shown in Figure 2 is based on the use of excess air as a cooling medium and of oil as fuel.

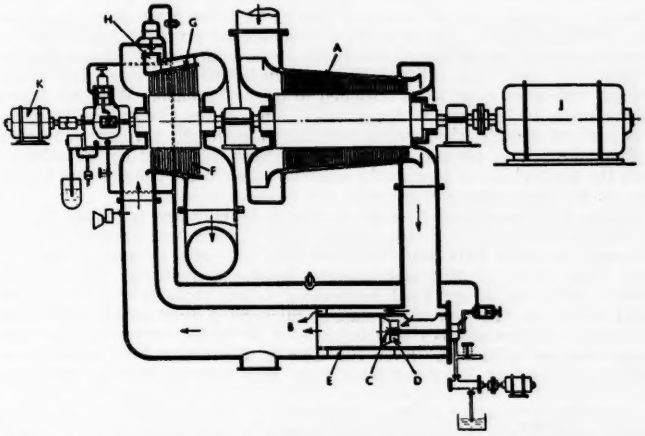


FIGURE 2.—DIAGRAM OF THE SIMPLEST FORM OF COMBUSTION TURBINE PLANT. With Reaction-Type Gas Turbine and Axial Compressor for Oil Fuel, and With Excess Air Cooling.

The total quantity of air (combustion plus cooling air) is drawn from the surrounding atmosphere by the axial compressor A, compressed to the combustion pressure of the order of 20-30 pounds per square inch gauge, and forced into the combustion chamber B. Part of the air serves as combustion air for the oil which enters the burner D at C, while the remainder is forced through the annular space between the wall of the combustion chamber and the burner jacket. Here it takes up heat from, and incidentally cools, the burner jacket, after which it mixes with the products of combustion, reducing their temperature to that admissible for the gas turbine blading F. In this simplest form of gas turbine, the gases go straight from the turbine G to the chimney. The speed of the gas turbine is governed by controlling the fuel oil supply, and by means of a by-pass valve H, which acts as safety valve. In addition to the compressor referred to above, the gas turbine drives the generator J either directly or through gearing. The only auxiliary machines of the plant are the starting motor K, which is of a size to ensure the compressor supplying enough air to the combustion chamber for lighting up, and the lubricating and fuel oil pumps.

It will be seen that the early inventors' aim to simplify the steam plant has been attained, there being no boiler with auxiliary equipment such as feed water pumps, water treatment plant, etc., no condenser with circulating water, air extraction, and condensate pumps, no water supply system, cooling towers or such like. The boiler is replaced by a simple combustion chamber, although

the compressor is now larger and more expensive than the forced and induced draught fans of a normal boiler installation.

Later inventors and scientific workers appear to have been striving for an improvement over the efficiency of the steam process, for, in about 1850, Redtenbacher wrote to Zeuner: "The fundamental principle of the generation and use of steam is wrong. It is to be hoped that steam engines will disappear in a not far distant future, as soon as we know more about the nature and effects of heat." The fact that Redtenbacher's prophecy has so far not come true, must be attributed mainly to the invention of the steam turbine, which gave a new lease of life to the steam cycle, and for a long while caused the gas turbine problem to be relegated to the background. With the advent of the steam turbine, one of the principal aims of the gas turbine inventors was realized, namely, the direct generation of power by a rotary movement without the help of pistons, cranks, etc., whereas this could not be achieved with the gas turbine, as long as the machine required for compressing the air was of the reciprocating type.

As far as is known, the first gas turbine to be built and tried was the hot air turbine developed by Stolze, which is said to have been designed in 1872, although the trials were made between 1900 and 1904 (Figures 3 and 4). This turbo set is of particular interest, because a multistage reaction gas turbine and a multistage axial compressor (most probably the first of its kind) were used. In view of the state of engineering development and limited knowledge of aerodynamics at that time, it is hardly surprising that this design was not a success. Even Sir Charles Parsons had to abandon the axial compressor at about the same time, after having done an enormous

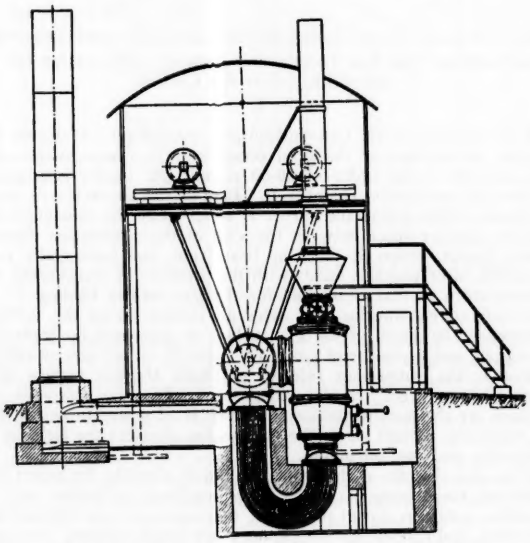


FIGURE 3.—STOLZE'S HOT AIR TURBINE (1900-4).

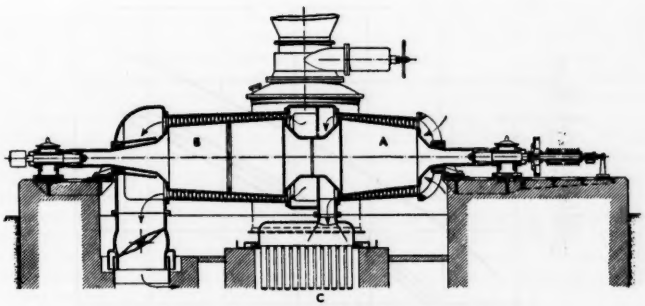


FIGURE 4.—STOLZE'S HOT AIR TURBINE.
A Multistage Axial Compressor. B Multistage Reaction Turbine.
C Heat Exchanger.

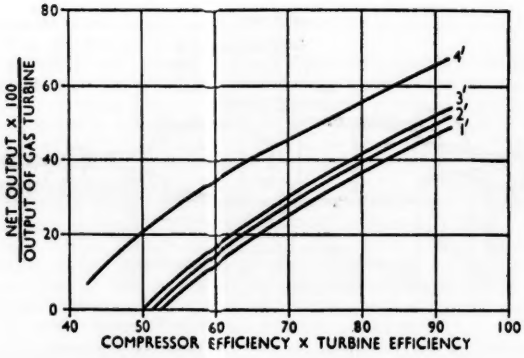
amount of work in connection with it, this being one of the few cases in which this great inventor, in spite of his engineering genius, and of the resources at his disposal, was compelled to give up a quest on which he had embarked.*

It is a strange coincidence that the first centrifugal compressor ever built (a Brown, Boveri three-cylinder turbo-compressor with 25 impellers, working all in series, to develop a pressure of 60 pounds per square inch absolute) was for a gas turbine designed by Armangaud Lemale which used paraffin oil as fuel. With a temperature of about 1030 degrees F. at the nozzles, obtained by injecting water into the combustion chamber, the turbine was just able to supply the power required for compressing the air. The net output was nil, and the efficiency was zero. The Société des Turbo-Moteurs, which built this turbine, did, however, reap some reward for their efforts, as the experience gained enabled them to improve considerably the performance of torpedoes by injecting and burning paraffin oil in the compressed air actuating the propelling engine. The conditions were here extremely favorable, as it was easily possible to avoid excessive temperatures, owing to the availability of an abundance of water for cooling purposes.

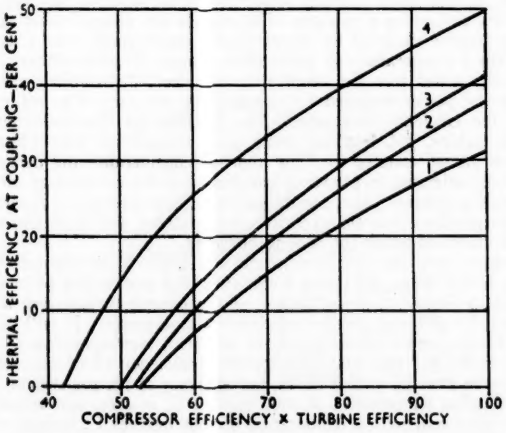
The reason why the first turbine did not prove a success can be seen in Figure 5, which shows the thermal efficiency of a gas turbine plant of the type illustrated in Figure 2, as a function of the overall turbine and blower efficiency, and for different turbine admission temperatures. It will be seen that even with an overall efficiency of 53 per cent, corresponding to a turbine efficiency of 78 per cent and a compressor efficiency of 68 per cent (values unlikely to be attained at that time), the efficiency of the gas turbine cycle with a gas admission temperature of 1000 degrees F. is zero, and would not have exceeded 5 per cent with a temperature of 1500 degrees F., a value which even now is still inadmissible. The cause of this unhappy result must be sought in the great volume of compressed air required for mixing with the combustion gases to reduce the combustion temperature of approximately 3300-3600 degrees F. to the value admissible for the gas turbine blading.

To get over this difficulty, which was thought insurmountable at the time, Holzwarth, in 1905, resorted to the explosion or constant-volume turbine.

* Stoney, Dr. G., Engineering, 1937, vol. 144, p. 695, "Scientific Activities of the late Hon. Sir Charles A. Parsons, O.M., F.R.S."



1′ $\frac{\text{Net output} \times 100}{\text{Output of gas turbine}}$ at 1,000 deg. F. before the turbine
2′ " " 1,200 " " "
3′ " " 1,500 " " "
4′ " " 2,200 " " "



1 Thermal efficiency at coupling of turbine at 1,000 deg. F.
2 " " " " 1,200 "
3 " " " " 1,500 "
4 " " " " 2,200 "

FIGURE 5.—THERMAL EFFICIENCY AT COUPLING OF GAS TURBINE AS A FUNCTION OF THE OVERALL EFFICIENCY OF COMPRESSOR AND TURBINE. For Different Temperatures at the Turbine Inlet Without Recuperation of Exhaust Heat

$$\text{Thermal Efficiency} = \frac{\text{Heat Equivalent of Output at Coupling}}{\text{Heat Supplied by Fuel}}.$$

In this turbine, the fuel (oil, blast furnace gas, and pulverized coal have so far been used) is fed to a closed combustion chamber filled with compressed air, and the mixture exploded, thus causing the pressure to rise to several times (approximately 4½ times) its original value. The combustion chambers, nozzles, impeller, and blades are water-cooled. The output of the compressor is only a fraction of that required for the combustion turbine, and a poor efficiency of the compressor has no longer such a disastrous effect, because (1) only a small excess of air is necessary for combustion, since water is used for cooling; and (2) the air need only be compressed to approximately one-quarter of the final explosion pressure.

These advantages are, however, accompanied by a considerable complication and increase in cost of the plant. First, a number of automatically operated valves is necessary for the combustion chamber; and second, to recover the heat given up to the cooling water, the latter must be evaporated and the steam used to actuate a turbine for driving the compressor.

In order that this turbine may be able to develop the power required by the compressor, it must run condensing, for which purpose a condenser with all its auxiliaries, and a cooling-water plant are needed. The first gas turbine of this type was built between 1906 and 1908 by Messrs. Körting of Hanover

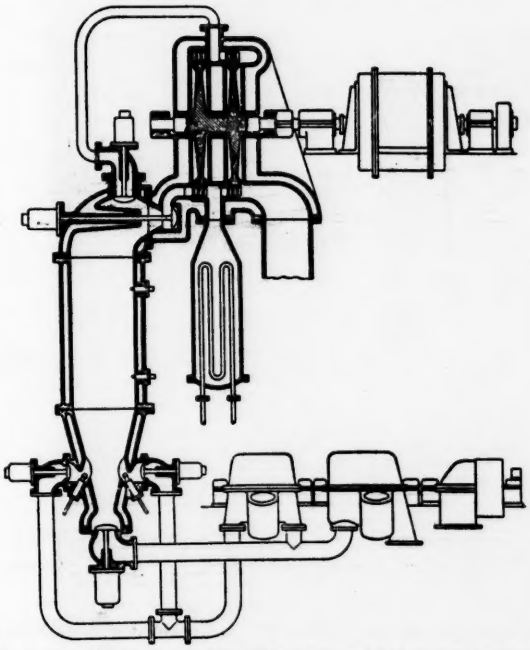


FIGURE 6.—DIAGRAM OF THE HOLZWARTH EXPLOSION TURBINE.
Two-Chamber Two-Stroke Design.

to the design of Dr. Holzwarth. On the basis of the results obtained with this experimental turbine, Messrs. Brown, Boveri built and tested between 1909 and 1913, to the order of Dr. Holzwarth, a second gas turbine with a nominal rating of 1000 HP, which, however, gave a net output of only about 200 HP. Several further Holzwarth gas turbines were built by Messrs. Thyssen during the period 1914 to 1927, but none of them was ever put into continuous operation. In 1928, Messrs. Brown, Boveri again took up the manufacture of a Holzwarth gas turbine and proposed for it a cycle which might be called the two-chamber, two-stroke cycle. The diagrammatic arrangement of the plant is shown in Figure 6, while Figure 7, Plate 1, is a view of the turbine on site. This unit is installed in a German steel plant, where it has been in operation with blast furnace gas since 1933. The results obtained have led to an order being placed for a larger turbine of 5000 HP, now being built by Messrs. Brown, Boveri at Mannheim.

The explosion turbine is really outside the scope of the present paper, and it is accordingly impossible to deal here more fully with the Holzwarth turbine, but it would be unfair not to acknowledge the enormous amount of work

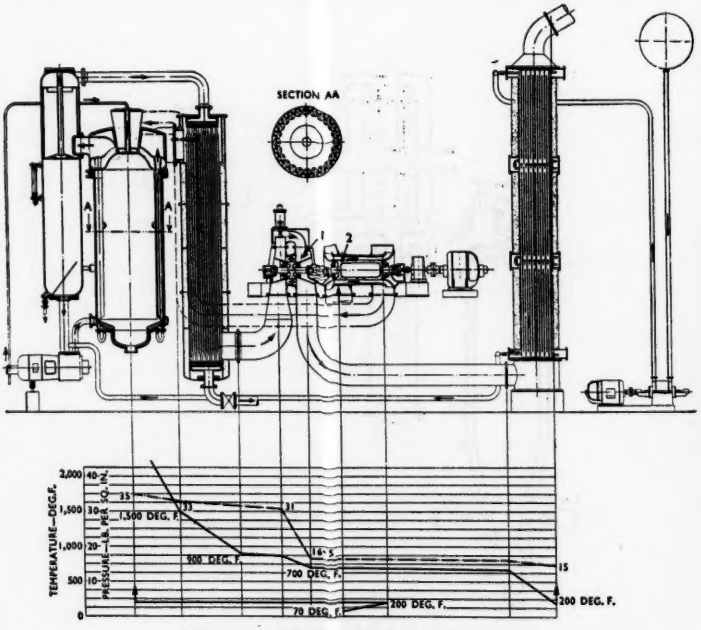


FIGURE 8.—VELOX STEAM GENERATOR.
A Highly Efficient Gas-Turbine-Driven Compressor Was Developed as an Auxiliary.
1 Gas Turbine. 2 Compressor.

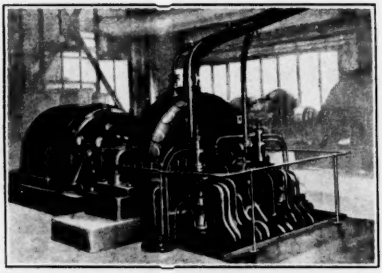


FIGURE 7.—HOLZWARTH EXPLOSION TURBINE WORKING WITH BLAST FURNACE GAS.
(Thyssen Steelworks, Hamborn.)

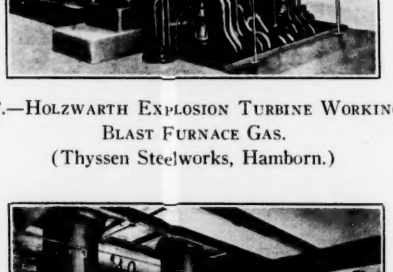


FIGURE 9.—GAS TURBINE SUPERCHARGING PLANT FOR THE HOUDRY OIL REFINERY PROCESS.
(Sun Oil Company, Marcus Hook, Near Philadelphia, U. S. A.)

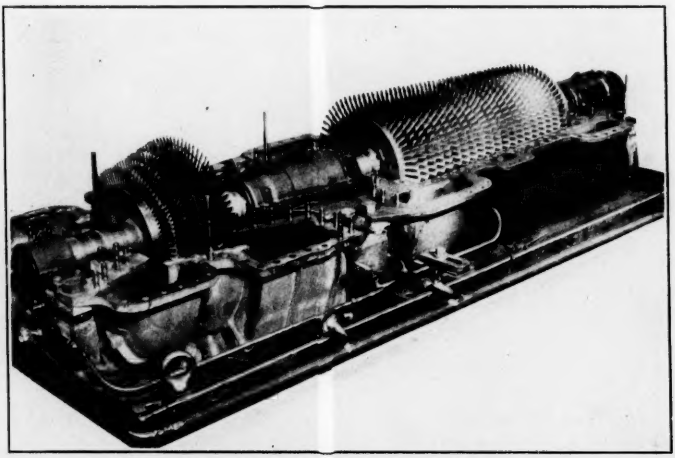


FIGURE 10.—GAS TURBINE COMPRESSOR UNIT SHOWN IN FIGURE 9, OPEN.

contributed during thirty years of unrelenting effort by Dr. Holzwarth and his collaborators, to the development of the gas turbine, and not to say a word of praise for their tenacity and the material sacrifices made by them in following up this idea.

The work done by Messrs. Brown, Boveri in connection with the Holzwarth gas turbine resulted in the development of the Velox boiler, whose principle is now well known to most engineers, but which nevertheless is referred to again here (Figure 8), because the experience gained with it led back to the combustion turbine. The Velox steam generator is a boiler fired under pressure, the pressure being produced by a compressor driven by a gas turbine, actuated by the exhaust gases of the boiler. Part of the pressure produced in the compressor is used to maintain high gas velocities in the heat-transmitting parts of the boiler, thus ensuring high rates of heat transfer. The remainder of the pressure head is used to drive the gas turbine. This application of the gas turbine as an auxiliary rendered essential the creation of a compressor-set having a high efficiency, otherwise the exhaust gas turbine would be unable to develop the power required for driving the compressor, and the deficiency would have to be supplied by another source, which would have reflected seriously on the efficiency of the boiler.

The problem was solved by the development of a four- or five-stage reaction turbine and a ten-to twelve-stage axial compressor, the design taking into account the results of the latest research in the field of aerodynamics.

Present Stage of Development of the Combustion Turbine. The attainment of efficiencies of 70 per cent and over in large sets of this kind suggested that the possibilities of the combustion turbine cycle for a prime mover without the use of steam be once more investigated. Figure 5, which has already been used to explain why former designs failed to deliver any useful output, shows that with overall efficiencies of 70-75 per cent it is possible to obtain the following net output efficiencies of the gas turbine cycle:—

Gas Temperature at Turbine Inlet, Degrees F.	Cycle Efficiency, Per Cent
1000	15-18
1200	19-23
1500	22-26

The question was: What temperature could safely be employed without harm to the gas turbine blading? On the basis of experience obtained with a large number of Velox boilers, and from the operating records of hundreds of exhaust-gas turbines for Diesel engine superchaging units, a temperature of 1000 degrees F. was considered as absolutely safe for uncooled blades made of the available heat-resisting steel, due allowance being made for inevitable temporary fluctuations of temperature during governing operations.

The efficiency corresponding to this temperature, referred to the output at the coupling and the net calorific value of the fuel, and assuming an overall compressor and turbine efficiency of 73-75 per cent, readily obtainable with net outputs of 2000-8000 Kw. could, according to the curves in Figure 5, be expected to attain 17-18 per cent.

Applications. The next question was: What use could there be for a turbine of this efficiency? In raising this question, it must be remembered that before high pressures were introduced, 18 per cent was considered quite a satisfactory efficiency for a steam turbine of this size, but with high pressures and temperatures, regenerative heating of the condensate, air-preheating, and all the dodges in use in modern power stations, it is now possible to attain

for such outputs coupling efficiencies approximating to 25 per cent. To attempt to compete with the modern base-load steam turbine would, therefore, be useless.

Supercharging in Chemical Process. The demand for hot compressed air in connection with the Houdry cracking process * created the first interesting field of application for the gas turbine. Many chemical processes are favorably affected by an increase in pressure, either by intensifying the action of catalysts or otherwise assisting the reaction, and resulting in an improvement in quality or an increase in quantity of the product, or by enabling smaller and therefore cheaper apparatus to be used. In many such processes, supercharging is impossible in practice, owing to the cost of compressing the air, but if the gases resulting from the process can be expanded in a gas turbine, and the power thus produced is sufficient to drive the compressor, supercharging is in many cases both feasible and economical.

This applies to the Houdry cracking process, where the power generated by the exhaust gases in the gas turbine more than suffices for supplying that taken by the compressor, the excess being used for driving a generator coupled to the set through gearing. Although the pressure drop in the apparatus used for the cracking process results in the electrical energy obtained being considerably less than that which would be delivered by a pure gas turbine, the energy produced is sufficient to cover the power requirements of the cracking

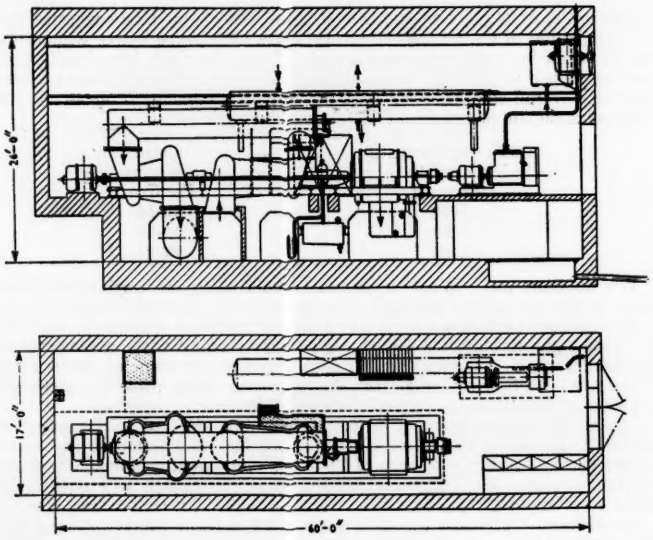


FIGURE 11.—BOMB-PROOF, GAS-TURBINE-DRIVEN, 4000-KW. STANDBY PLANT. City of Neuchâtel.

* See The Oil and Gas Journal, 1938, vol. 37, p. 40.

process, for which a Velox boiler supplies the necessary steam. The gas turbine has an output of 5300 Kw., and the input to the compressor is 4400 Kw., the excess of 900 Kw. being converted into electrical energy. The unit in question, shown in Figure 9, Plate 1, has been in operation day and night for more than two years in the Marcus Hook Refinery of the Sun Oil Company, near Philadelphia, U. S. A. Figure 10, Plate 2, shows the unit opened up, the gas turbine being on the left, and the multistage axial compressor on the right. The experience gained with this gas turbine resulted in orders for ten additional units, of which two are already running, one in Italy and the other in Texas.

Standby and Peak Load Plants. Although the combustion turbine cannot yet compete with the modern steam plant for base-load purposes, it deserves serious consideration as a standby and peak load unit, on account of its special advantages. It is of simple design, cheap, light, requires little space, and, last but not least, is not dependent on any water supply. This means that the foundations and buildings are cheap, and that the plant can be installed right at the point of consumption.

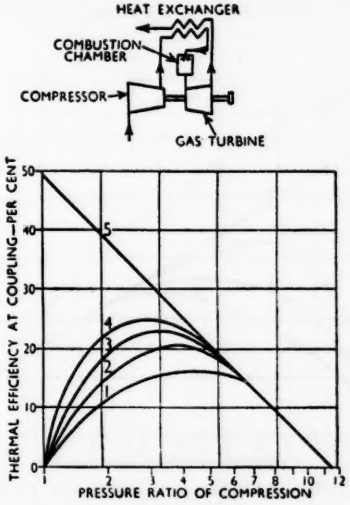
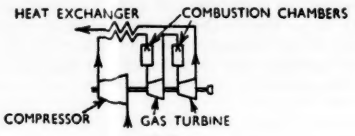


FIGURE 12.—EFFICIENCY AT COUPLING OF 2000-KW. COMBUSTION TURBINE.
With Temperature of 1000 Degrees F. at the Turbine Inlet, and Air Preheating by Exhaust Gases With Different Sizes of Preheater.
Surface of Heat Exchanger for 2000 Kw.:—

Curve	1	0	Square	Feet
"	2	5000	"	"
"	3	15,000	"	"
"	4	30,000	"	"
"	5	∞	"	" (Theoretical)

Temperature Before Turbine, 1000 Degrees F.; Temperature of Intake Air, 68 Degrees F.; Compression Without Cooling.

These advantages are of particular importance in the case of so-called bomb-proof emergency stations, such as are now frequently built to meet the power requirements of key industries in time of war, and for the maintenance of the essential services of towns (waterworks, etc.), which normally receive current over transmission lines, in the event of the power supply being interrupted. An example of this type of plant is the 4000-Kw. set ordered for the city of Neuchâtel some eight months ago. Figure 11 illustrates the extraordinary simplicity of such a power station. The set operates on the principle of the diagram shown in Figure 2. The only auxiliary plant is the separate small Diesel-driven alternator, which in the event of a complete failure of power serves to supply the starting motor of the gas turbine.



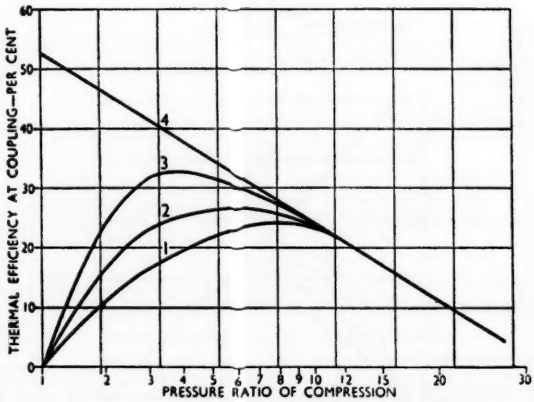


FIGURE 13.—EFFICIENCY AT COUPLING OF A 2000-KW. COMBUSTION TURBINE. With a Temperature of 1000 Degrees F. at the Turbine Inlet and Air Preheating by Exhaust Gases With Different Sizes of Preheater, and Reheating of the Gases to the Initial Temperature Between Two Stages.

Surface of Heat Exchanger for 2000 Kw.:—

Curve	1	0	Square Feet
"	2	5000	" "
"	3	15,000	" "
"	4	30,000	" "
"	5	∞	" "

Temperature Before Turbine, 1000 Degrees F.; Temperature of Intake Air, 68 Degrees F.; Compression Without Cooling. Turbine Efficiency, 86 Per Cent; Compressor Efficiency, 83 Per Cent.

Improvements of the Cycle. The applications referred to above correspond to the simplest design shown in Figure 2. An important improvement is attained by the utilization of the heat of the exhaust gases of the turbine to preheat the compressed air, as shown in Figure 12. Here the coupling efficiencies of the gas turbine set obtainable with heat exchangers of different sizes for an output of 2000 Kw., and a turbine admission temperature of 1000 degrees F. are plotted against the pressure ratio of the compressor. It will be seen from these curves that the recuperation of the heat of the exhaust gases in a heat exchanger of 5000 square feet raises the cycle efficiency from 16.5 to 21 per cent, i. e., brings about an improvement of 27 per cent.

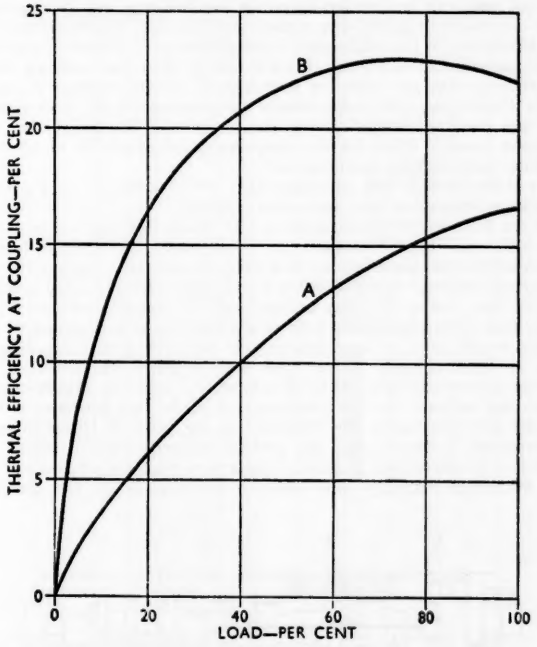


FIGURE 14.—THERMAL EFFICIENCY AT COUPLING AS A FUNCTION OF THE LOAD.

A Thermal Efficiency Without Recuperation; Single-Shaft Arrangement With One Turbine.

B Thermal Efficiency With Recuperation and Two-Shaft Arrangement With Two Turbines, One Driving the Compressor, the Other Driving the Generator.

Temperature Before Turbine, 1000 Degrees F.; Surface of Heat Exchanger, 7500 Square Feet for 2000 Kw.

A further considerable improvement can be achieved by dividing the gas turbine into several cylinders and reheating the gas between those cylinders to the initial temperature, as shown in Figure 13, which is drawn for a 2000-Kw. unit operating with an admission temperature of 1000 degrees F. and one intermediate reheat stage. As can be seen from this diagram, it is thus possible to obtain an efficiency of 19 per cent without preheating the combustion air, and 22½ per cent with the above-mentioned preheater of 5000 square feet. The number of reheating stages is limited by the increased cost of the plant, as well as by the pressure drop in the piping and combustion chambers.

A further innovation which improves the efficiency at fractional loads, but not at full load, can be obtained by using two turbines, one driving the compressor only without giving any power, and the other supplying only power.

The efficiencies of the single-shaft arrangement as in Figure 2, and the two-shaft arrangement as in Figure 11, for a unit of 2000 Kw. with air preheater, and operating with an admission temperature of 1000 degrees F., are compared in Figure 14. The improvement at fractional loads is brought about by the fact that the turbine driving the compressor can always be operated at the most suitable speed for the compressor, independently of the speed of the turbine supplying the useful power.

These improvements can advantageously be incorporated in the following proposed applications of the combustion turbine.

Proposed New Fields of Application. An interesting application of the gas turbine is the gas turbine locomotive, as an alternative to the steam locomotive. It is well known that the efficiency of a steam locomotive is only of the order of 8-12 per cent, the latter figure allowing for all improvements made in the course of the last few years. The gas turbine, with its coupling efficiency of 17-20 per cent, finds here a promising field of application, since it should be possible to obtain an efficiency of approximately 15 per cent with mechanical transmission, or approximately 14 per cent with electrical transmission. These efficiencies do not naturally allow of competition with the Diesel engine, but the difference between the fuel consumptions of the two machines is in many cases fully compensated by the difference in the price of Diesel oil and fuel oil. Moreover, in many cases the problem of getting more power from a given size of locomotive is of greater importance than the efficiency. A noteworthy advantage compared with steam locomotives is the fact that the gas

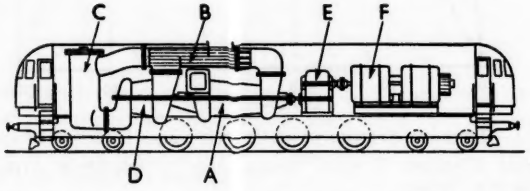


FIGURE 15.—DIAGRAM OF A GAS-TURBINE-ELECTRIC LOCOMOTIVE.

A Compressor.
B Heat Exchanger.
C Combustion Chamber.
D Gas Turbine.
E Gearbox.
F Generator.

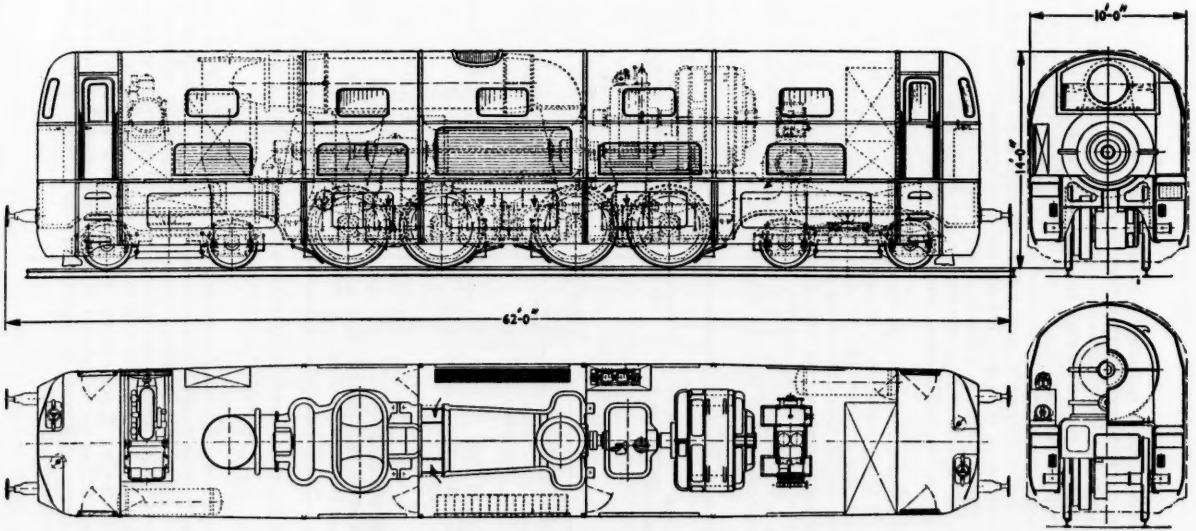


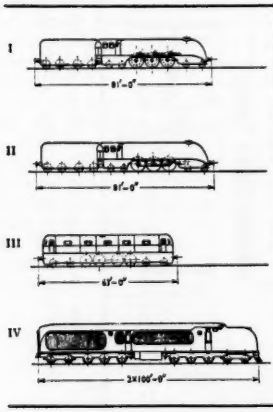
FIGURE 16.—GAS-TURBINE-ELECTRIC-DRIVEN LOCOMOTIVE.
3000 HP at Generator Coupling or 2500 HP at Wheel Tread.

turbine locomotive needs no water, thus doing away with cleaning of boilers and interruptions of service resulting therefrom.

In the case of electrical transmission, only the gas turbine part is new, the generators, motors, and switchgear being adapted from the Diesel-electric locomotive without modification. The gas turbine which drives the generator through gearing can always be operated at the most suitable speed for the compressor, so that the engine gives a high efficiency at all speeds and loads. Such a locomotive is, of course, heavier and more expensive than one with mechanical transmission, but difficulties may have to be overcome with the latter, especially in case of high outputs, since it will be necessary to provide in addition to the forward turbines, reverse turbines or corresponding reversing gears.

Figure 15 shows a gas turbine locomotive with electrical transmission; A is the blower, B an air preheater, C the combustion chamber, D the gas turbine driving the generator F through the gearing E; while Figure 17 compares different designs and sizes of locomotives.

Figure 16 shows a project for a gas-turbine-driven locomotive for an output of 3000 HP at the generator coupling, and an output of approximately 2500 HP at the wheel tread. The design of the locomotive and the main and auxiliary machines is clear from the references to the drawing.



Technical data	High-pressure locomotive with reciprocating engine	Standard locomotive with reciprocating engine	Gas-turbine locomotive with electrical transmission	High-pressure locomotive with condensing plant and electrical transmission
Locomotive .	I	II	III	IV
Railway system	North	North	Project	Union Pacific Railroad
Type . .	2 C_0 2	2 C 2	2 D_0 2	2 C_0–C_0 2
Builder .	Swiss Locomotive and Machine Works, Winterthur, and Alsacienne	Alsacienne	Brown, Boveri	General Electric Company
Power at rim of wheel, h.p.	3,000	3,000	3,000	(2) × 2,500
Thermal efficiency at rim of wheel, per cent	12	10	17	14·5
Service weight with tender, tons	194	180	135	(2) × 234
Speed, m.p.h. .	75	75	85	100
Fuel . .	Coal	Coal	Fuel oil	Fuel oil
Length including tender, feet	81	81	63	(2) × 100

FIGURE 17.—GAS TURBINE LOCOMOTIVE COMPARED WITH STEAM AND STEAM-ELECTRIC LOCOMOTIVES.

Ship Propulsion. A further promising field of application for the combustion turbine is to be found in the propulsion of ships. Although competing with modern geared turbine plants cannot yet be contemplated, the question of placing 2000-4000 HP combustion turbines instead of reciprocating engines on oil-fired ships already deserves full investigation by the interested parties, since efficiencies over 20 per cent can be obtained with a combustion turbine plant through utilization of the heat of the exhaust gases for preheating the air, which means an increase of approximately 20 per cent on those attained with reciprocating engines.

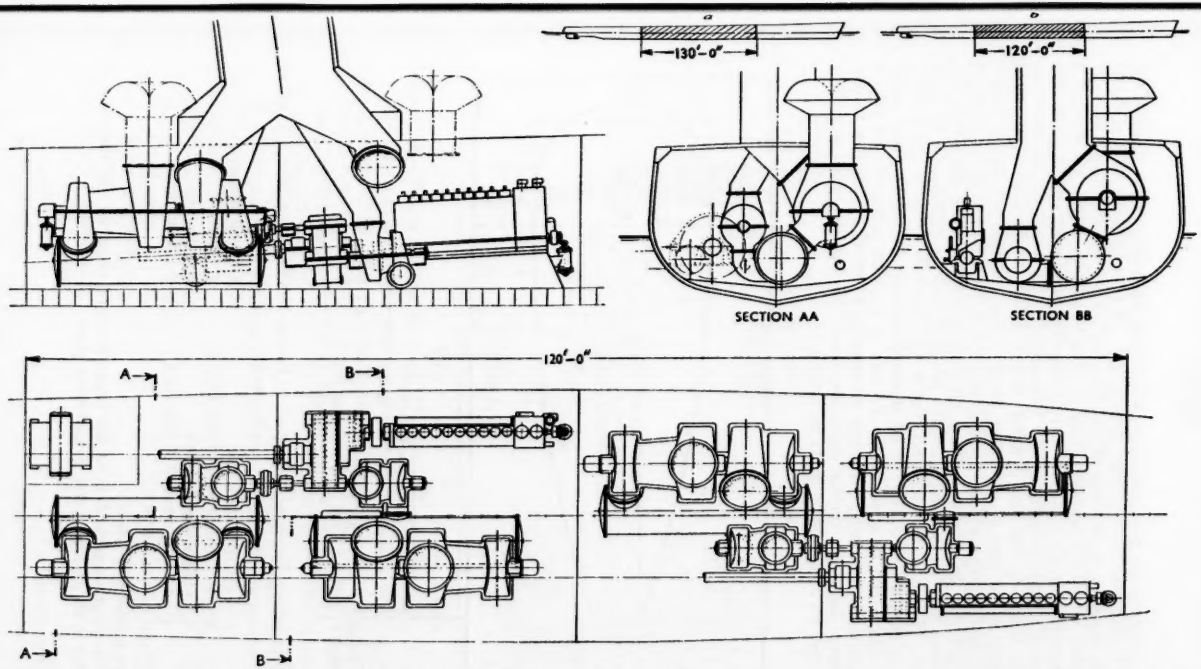


FIGURE 18.—COMPARISON BETWEEN A (2) × 16,000-SHP STEAM TURBINE-DRIVEN DESTROYER AND A (2) × 18,000-SHP COMBINED COMBUSTION TURBINE- AND DIESEL ENGINE-DRIVEN DESTROYER.

a (2) × 16,000-SHP Steam Turbine: Weight of Machine, 30 Pounds per SHP; Fuel Consumption at Cruising Speed, 1.12 Pounds per SHP; at Maximum Output, 0.86 Pounds per SHP.

b (2) × 18,000-SHP Gas Turbine: Weight of Machine, 20 Pounds per SHP; Fuel Consumption at Cruising Speed, 0.4 Pounds per SHP; at Maximum Output, 0.84 Pounds per SHP.

The prospects for merchant ships will depend upon the conditions applying, and each case must be examined individually. A very different state of things prevails, however, in the case of warships, and in particular torpedo boats or destroyers, which are usually driven by steam turbines. Here the drive is a compromise, since the top speed entails outputs ten or even twenty times that required at normal cruising speed. Consequently, the steam and oil consumptions at both the top and cruising speeds differ considerably from those customary in the case of stationary plants. The efficiency of such a marine turbine plant at full speed is approximately 14-18 per cent, i. e., of the order of that which can be obtained with a gas turbine plant without preheating the combustion air. At cruising speed the efficiency may be even as low as 11-14 per cent. There is, therefore, no reason why a simple form of gas turbine, which in regard to the average oil consumption over the whole speed range of the ship would be equal to the average of a steam drive, should not be considered. If, however, the many advantages afforded by the gas turbine drive over the steam turbine drive are taken into account, even a slightly higher fuel oil consumption could be accepted.

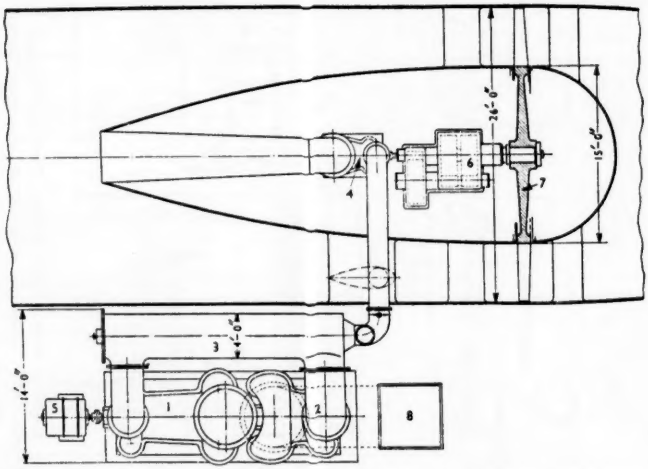


FIGURE 19.—GAS TURBINE PLANT AS A PRIME MOVER FOR A WIND TUNNEL BLOWER.

1 Axial-Flow Blower.
2 Gas Turbine for Driving Item 1.
3 Combustion Chamber.
4 Gas Turbine for Driving Item 7.
5 Pony Motor.
6 Gear.
7 Fan.
8 Chimney.

These advantages are:—

1. The great simplification of the plant, brought about by the omission of the boiler with its feed water evaporating plant.

2. Reduced vulnerability when in action, due to the absence of large volumes of water and steam stored in boilers and steam pipes.

3. The simplification due to the omission of the condensing plant with all its auxiliaries, corrosion problems, and other troubles.

4. The rapidity with which the gas turbine plant can be brought into service, only as many seconds being required as minutes are needed for a boiler installation.

5. The fact that the plant stops and becomes cool within a few seconds of cutting off the fuel supply, because the stored energy is quickly absorbed by the compressor in pumping cold air through, and thus cooling, the combustion chamber and gas turbine.

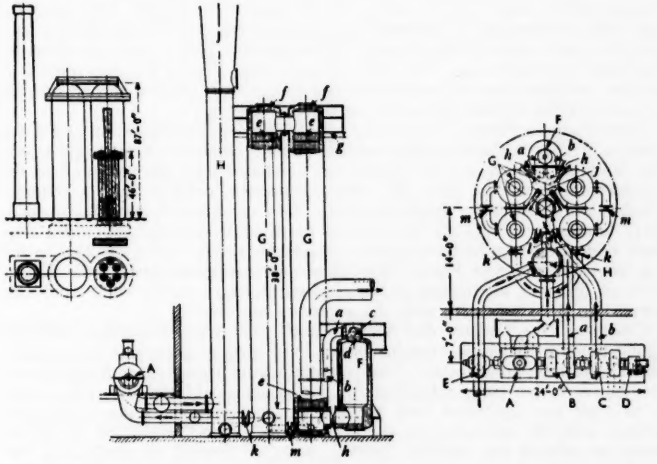


FIGURE 20.—VELOX BLAST HEATER, WITH COMBUSTION UNDER PRESSURE SUPERCHARGED BY GAS-TURBINE-DRIVEN AIR AND GAS COMPRESSORS.

Output, 25,000 Cubic Feet per Minute; Blast Temperature, 1650 Degrees F.; Combustion Pressure, 22 Pounds per Square Inch Gauge; Duration of Cycle, 10 Minutes.

Supercharging Set:

A Gas Turbine.
B Combustion Air Compressor.
C Gas Compressor.
D Auxiliary Motor.
E Booster Blower for the Blast Air.

Blast Heater:

F Combustion Chamber.
a Combustion Air Supply.
b Gas Supply.
c Mixing Chamber.
d Burner.
G Regenerators.
e Refractory.
H Air Preheater for the Blast (Recuperation).
J Stack.

Change-Over Valve Gear:

h Hot Gases.
j Hot Blast.
k Cold Gases.
l Cold Blast.
m By-pass for Temperature Regulator.

A further appreciable improvement is obtained if Diesel engine drive is resorted to for the cruising speed. This can be done because the gas turbine is much lighter and requires less space, so that the complete plant—turbine plus Diesel engine—is still more compact and lighter than a steam turbine plant. This is shown by Figure 18, which compares a plant comprising two 16,000-SHP steam turbines with a $2 \times 18{,}000$-SHP combustion turbine and Diesel engine plant.

Wind Tunnels. A comparatively small but interesting field of application is the drive of wind tunnel blowers for testing aeroplane forms. Such drives, which require a considerable amount of power (up to 5000 HP) for testing purposes, and, therefore for short periods only, are very unfavored consumers of current, and are charged for energy at high rates; moreover, the periods during which power can be taken are frequently restricted. Here also the gas turbine drive can be resorted to, as, on account of its relatively low cost and the small masses which have to be heated up, it is more suitable than, for instance, a steam plant. As will be seen from Figure 19, the turbine is located outside the tunnel and drives the compressor, while a second turbine, arranged in the tunnel, drives the fan through gearing.

Blast Furnace Plants. Interesting possibilities for the application of the gas turbine drive are offered by blast furnace plants, especially when the air supply for the blast furnace and its heating are combined with the operation of the combustion turbine by using the same compressor, together with a common combustion chamber for both purposes. This combination becomes particularly interesting if the air heaters are fired under pressure, in which case their dimensions can be considerably reduced, as shown in Figure 20. In the top left-hand corner of Figure 20 is shown a comparison between the dimensions of ordinary air heaters and supercharged ones; on the right-hand side are shown the air heaters with the accessory machines.

Combined Gas Turbine and Steam Plant. The more important fields of application for which gas turbine plants have already been supplied or proposed have been enumerated. The potentialities of this type of prime mover are, however, far from exhausted by this account. Mention may be made of at least one additional field, namely the combination of the combustion turbine with the utilization of the heat of the exhaust gases for producing steam in exhaust gas boilers, together with, or instead of, preheating the combustion air. In many cases where there is a demand for steam in addition to energy, the combustion turbine with exhaust gas boiler is more economical and cheaper than a high-pressure steam plant with back-pressure or extraction turbines.

Glimpse Into the Future. Figure 5 shows that an increase in the overall efficiency of compressor and turbine from 70 to 75 per cent raises the cycle efficiency from 15 to 18 per cent, representing an improvement of 20 per cent. Every 1 per cent increase in the efficiency of the compressor or the turbine means, therefore, an improvement of 4 per cent in the cycle efficiency. Here, then, is a field from which much progress may be expected.

Figure 5 also shows the improvement accruing from the use of higher temperatures at the gas turbine inlet. Such temperatures will be admissible either when materials of a correspondingly high creep strength are available, or when some means is devised for protecting the blading of the materials now used against the effects of such high temperatures. It can be seen from the curves that with an overall compressor and turbine efficiency of 76 per cent, increasing the temperature at the turbine inlet from 1000 to 1200 degrees F. raises the cycle efficiency from 18 to 23 per cent, representing an improvement of 28 per cent. Figure 21, which gives the cycle efficiency for a two-cylinder gas turbine of 2000 Kw. with intermediate reheating and a temperature of

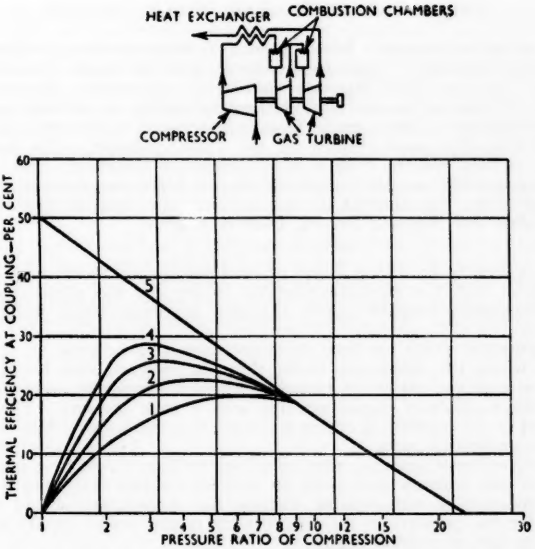


FIGURE 21.—EFFICIENCY AT COUPLING OF A 2000-KW. COMBUSTION TURBINE.

Temperature, 1200 Degrees F. at Turbine Inlet; Preheating of the Air by Exhaust Gases With Different Sizes of Preheaters; Reheating of the Gases to the Initial Temperature Between Two Stages.

Surface of Heat Exchanger for 2000 Kw.:—

Curve	1	0	Square Feet	
"	2	5000	"	"
"	3	30,000	"	"
"	4	∞	"	" (Theoretical)

Temperature Before Turbine, 1200 Degrees F.; Temperature of Intake Air, 68 Degrees F.; Turbine Efficiency, 86 Per Cent; Compressor Efficiency, 83 Per Cent.

1200 degrees F., shows that the efficiency attains 24 per cent without air preheating, 26.5 with a preheater of 5000 square feet heating surface, and 33 per cent with a preheater of 30,000 square feet heating surface.

The prospects of the gas turbine, if it is possible—as the author believes it is—to raise the temperature in the near future to 1200 degrees F., can be readily appreciated. This belief is based upon the experience obtained with a number of Diesel engine supercharging units which have been in operation for a considerable time at temperatures approaching this value.

The picture which this paper has attempted to convey, of the history, present state, and future of the combustion turbine, is necessarily incomplete, but it will serve to show that the subject is a promising one, full of interesting possibilities.

THE GÖTAVERKEN PNEUMATIC ENGINE.

The following abstract is from a paper "Thermodynamics of a New Type of Marine Machinery: Combustion Engines with Pneumatic Power Transmission," by Dr. H. G. Hammar and Mr. E. Johansson. It describes a method of producing an operating medium by mixing the exhaust gas from a Diesel cylinder working under high back pressure with cooling air compressed from the power output of the Diesel cylinder. The operating medium is then used to drive a piston engine or a turbine. The original paper was read before the Institute of Marine Engineers, London, England, April 18, 1939. The abstract is reprinted from the April 20, 1939, issue of Shipbuilding and Shipping Record, London, England.

With a view to producing a type of machinery which would have the advantages of both the internal-combustion engine and the steam engine, an internal-combustion engine system has been developed which will now be described.

The principle which has been developed aims at generating an operating medium which, like steam, can be fed through pipes and used for operating either piston-driven engines or turbines. The driving medium is produced by an internal-combustion engine working with a back pressure. The power developed in the combustion cylinders is used to compress air which is mixed with the combustion gases.

It was considered necessary that the driving medium should have a temperature such that ordinary steel could be used for the pipe lines, and if it was to be employed in reciprocating engines, the temperature should be low enough for the cylinders of the reciprocating engine not to require cooling; similar, in fact, to a steam engine.

The temperature of the driving medium was therefore established normally at 425 degrees C., with maximum of 450 degrees C., for reciprocating machinery and 500 degrees C. for turbines.

The investigations which were carried out in order to determine the pressure which the driving medium would reach if the temperature was established at 425 degrees C. indicated that the pressure became low if the normal combustion pressure of the Diesel motor was used. It was, therefore, found necessary to increase the combustion pressure. This caused no difficulty. On the contrary, it turned out to be an advantage, because a greater expansion ratio, with consequent improvement of the thermal efficiency, was obtained together with supercharge of the combustion cylinders.

Investigation showed that if a combustion pressure of 60 atmospheres be selected with air injection and 70 atmospheres with mechanical injection of the fuel, the pressure of the driving medium becomes 5 atmospheres or 4.0 kg./cm.^{2} above atmospheric pressure.

Investigation also showed that the thermal efficiency obtained with this system was quite as high as that obtained with a Diesel engine. This is due to the fact that the indicated power obtained when the combustion gases are permitted to expand from the increased combustion pressure adopted in this system to atmospheric pressure, is much greater than that obtained with a Diesel engine, where the expansion of the gases is cut off at 3 atmospheres.

The combustion engine for producing the gas mixture, which might be called the generator, is a fast running 2-stroke cycle engine with a high supercharge, and an effective mean pressure of 7.5 to 8.0 kg./cm.^{2} This engine, therefore, becomes very light compared with a Diesel engine intended for direct propeller drive.

It is therefore possible to build large engine plants, in which the weight of the generators and the propelling machinery will not exceed 20 kg. per SHP. For small engine plants the weight can be kept very low.

The generator performs the same function as the steam boiler in a steam engine plant and has therefore been popularly named a "Diesel boiler."

The first experiments with machinery of the type described were carried out during the years 1927 to 1929. The machinery comprised a one-cylinder, 2-stroke cycle, single-acting generator combined with a 2-cylinder double-acting reciprocating engine, fitted with poppet valves for gas inlet and exhaust. The combustion cylinder of the generator had a bore of 350 millimeters by 520-millimeter stroke. For the reciprocating engine both the bore and the stroke were 500 millimeters.

A fuel consumption of 189 grammes (0.416 pound) per EHP per hour was obtained with this machinery, which corresponds to a thermal efficiency of 33.5 per cent.

Figures Nos. 1 and 2 show indicator diagrams taken from this machinery.

The next step in the development of the system was the building of a tugboat with a 300 SHP propelling engine with pneumatic power transmission. This vessel was built in 1931 and fitted with machinery comprising a two-cylinder, 2-stroke cycle, single-acting generator with combustion cylinders

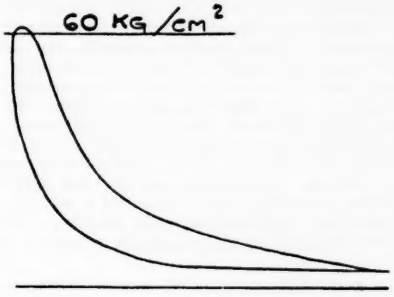


FIGURE 1.

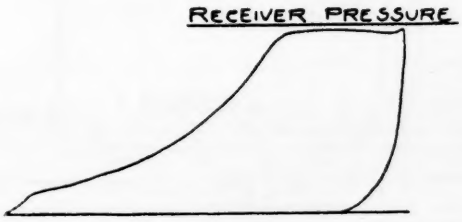


FIGURE 2.

having a bore of 300 millimeters by 480-millimeter stroke, and the above-mentioned reciprocating engine with a bore and a stroke of 500 millimeters, built for the experimental engine plant. The quantity of gas required for 300 SHP is produced by the generator when running at 280 RPM. The propelling engine is capable of developing 300 SHP at revolutions varying between 127 and 172 per minute, depending on the speed of the boat.

The engine is fitted with Klugs valve gear, which facilitates very quick maneuvering. The control of the power and the changing of the direction of movement is effected by a lever. When the lever is in the mid-position the engine is stopped, and when it is moved in the one or the other direction the engine goes ahead or astern. The power is regulated by the distance of the lever from the mid-position. The pull of the tug on a fixed object is 5 tons. This boat has been in service ever since it was built and has provided the most valuable experience. One observation which is most interesting is that the wear in the cylinders of the propelling engine is very small. It was thought that the wear would be great on account of the high temperature of the gas, but on the contrary it has become evident that the wear is small; after nearly eight years' service it amounts to only about 0.2 millimeters.

After experience had been gained in the tugboat of machinery of the foregoing type, the research and patrol vessel *Skagerak* was built. The stipulated requirements for this vessel were as follows:

The vessel should be capable of running at speeds ranging from 1 knot to full speed and be equipped for trawling and ice-breaking. The normal speed should be 11¼ knots. When ice-breaking the propelling machinery should develop at least 850 SHP.

Owing to these special requirements, it was considered advisable to fit two generators in the vessel. When running normally, one generator only is used, but at full speed, and when ice-breaking, both generators are used. All deck machinery, such as steering gear, windlass, capstan and trawl winch are pneumatically driven. The propelling engine is fitted with three cylinders having dimensions of 600 millimeter bore by 720 millimeter stroke. During the trial trip the engine developed over 1000 SHP at 141 RPM. The generators, each having two cylnders with a bore of 370 millimeters by 480 millimeter stroke, generate the required gas for 1000 SHP at 300 RPM. The fuel consumption at normal speed using one generator amounts to 192 grams per SHP per hour, including auxiliary machinery, which corresponds to about 185 grams per SHP per hour for the propelling machinery alone

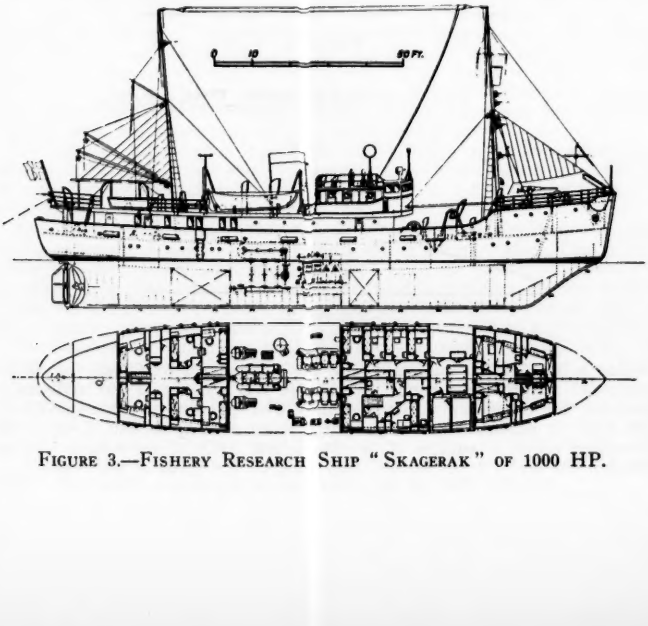


FIGURE 3.—FISHERY RESEARCH SHIP "SKAGERAK" OF 1000 HP.

and a thermal efficiency of 34.2 per cent. The number of revolutions of the propeller at normal speed is 110 per minute. The vessel was completed during the summer of 1935 and has thus been in service for four years.

Figure 3 shows a drawing of the vessel.

A further advance in the development of the system of internal-combustion engines with pneumatic power transmission is represented by machinery of 6500 SHP, which is in course of construction. Here the propelling engines consist of turbines. This machinery is intended to replace the steam turbine plant at present fitted in a mine-laying vessel. The machinery comprises four 6-cylinder generators with a bore of 370 millimeters by 480 millimeter stroke and two turbine units each coupled to one propeller shaft by means of single reduction gears. Each turbine unit comprises ahead and astern turbines. The ahead turbines are fitted with regulating valves for ¼, ½ and full load. Three of the four generators are completed and have been tested in the shop in combination with the turbines which are of De Laval manufacture. The machinery will be installed in the vessel during the coming summer. As the boilers at present are coal-fired, the weight of the fuel consumed by the new machinery will be only one-fifth that of the existing machinery. A plan of the machinery room and the boiler room is shown in Figure 4.

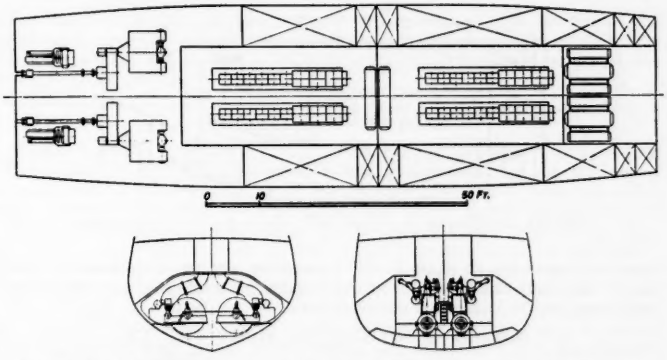


FIGURE 4.—MACHINERY LAYOUT FOR THE CONVERSION OF A MINELAYER.

A proposal for a small icebreaker intended for canals and lakes is shown in Figure 5. This icebreaker is intended to have forward and aft propellers and motor machinery of 900 SHP having pneumatic power transmission. The normal power is 600 SHP for the aft engine and 300 SHP for the forward engine, but, if desired, the whole power can be utilized in the aft engine. The propelling machinery comprises two generators, each with four combustion cylinders with a bore of 185 millimeters and two propelling engines, the aft one with four cylinders and the forward one with two cylnders 500 millimeter bore by 500 millimeter stroke. The generators will produce the quantity of gas required for full power at 530 RPM. Maneuvering of the propelling machinery is carried out from the bridge. Windlass and steering engine are pneumatically driven.

The weight of the whole machinery, including auxiliary engines, shafts, propellers, pipe lines and all mechanical equipment, is estimated at 97 tons.

One of the factors which, in addition to the suitability of the form of the vessel, is of the greatest importance in determining the ability of the ice-

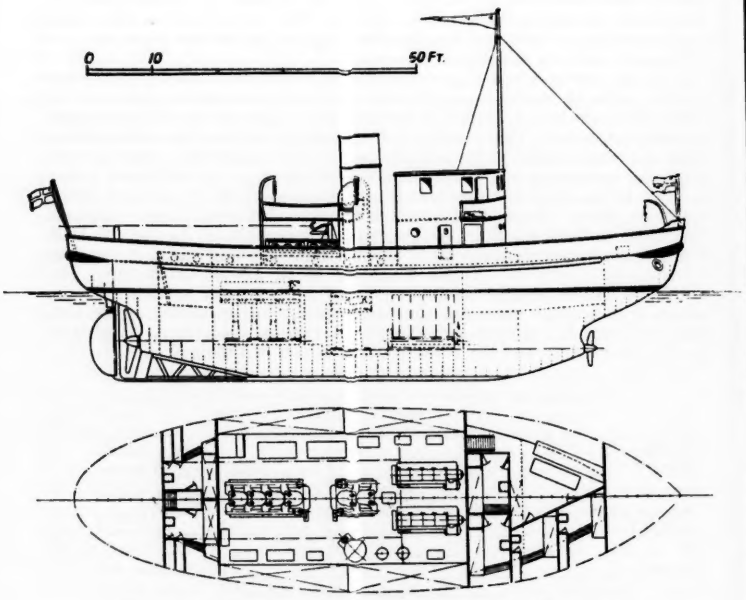


FIGURE 5.—PROPOSAL OF 900 HP FOR A SMALL ICEBREAKER.

breaker to perform its duty, is the horsepower per ton displacement. This figure has been calculated for various types of machinery and the undermentioned results have been obtained, viz.:

ICEBREAKER OF 900 SHP MACHINERY (HP/TΔ).

	SHP per ton	IHP per ton
Icebreaker with—		
Steam engine	1.88	2.14
Diesel-electric machinery	2.28	—
Direct-coupled Diesel motors	2.69	—
Pneumatic power transmission	3.21	—

200,000 SHP TURBINE MACHINERY.

In order to obtain an engine plant developing this power a large number of generators must be used as the diameter of the combustion cylinders, with regard to weight of the machinery and heat stresses, cannot be made too large. For the machinery proposed, the bore of the combustion cylinders is 370 millimeters and 32 generators will be required, each with 10 cylinders, in order to generate the necessary quantity of gas. The number of generators required will thus be about the same as the number of steam boilers required

for a steam plant. When utilizing the above-mentioned power for the propulsion of a ship, four propellers must be employed. In order to avoid too large dimensions of the turbine gears it is suitable to drive each propeller with four turbines. The machinery, which is shown in Figure 6, therefore

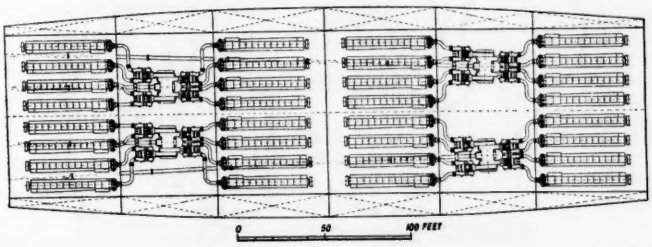


FIGURE 6.—GENERATOR SETS FOR TRANSATLANTIC LINER WITH 200,000 HP PNEUMATIC POWER TRANSMISSION.

comprises 4 propellers, 16 turbines and 32 generators. The weight of the generators, turbines and gearing included in the machinery is estimated to be about 4500 tons. If steel be used to a greater extent than estimated, the weight can be reduced considerably.

GENERATOR.

Figure 7 shows a section through the combustion cylinder of a generator. The cylinder is provided with ports in the cylinder wall for the scavenging air and with a valve in the top for the exhaust gases. The principal difference between the cylinder for a generator and that of a 2-stroke cycle Diesel engine is that with a Diesel engine the exhaust is closed at a certain fixed position of the crankshaft, while with a generator the timing of the closing point is varied with the gas pressure. This is necessary in order to enable starting and running of the generator at low gas pressure and also to prevent too high a compression and combustion pressure when running at high gas pressure. The closing point of the valve can be timed in such a manner that the compression period when starting and running with a gas pressure below 0.5 kg./cm.^{2} is 85 per cent of the stroke, while at a gas pressure of 3 kg./cm.^{2} or more the compression period is about 40 per cent of the stroke only. See Figure 8.

The mechanical means for operating the exhaust valve and for timing the closing point consists of one fixed and one adjustable cam on the camshaft. The fixed cam A opens the valve and the adjustable cam B determines the closing point. The adjustable cam, which is loose on the camshaft, is provided with a gear and is driven by another gear, with the same number of teeth, which is keyed on to the shaft below the camshaft. The lower shaft is driven by a gear from the camshaft in such a manner that the lower shaft can be turned in relation to the camshaft by shifting the axial position of the gear. When shifting this gear axially the adjustable cams are thus turned on the camshaft. The turning of this shaft takes place automatically by means of a piston operated by the pressure of the gas mixture.

At start and when running at low gas pressure cam B is located behind cam A, but as the gas pressure increases cam B moves in the opposite direction to the rotation of the camshaft, thereby delaying the closing of the exhaust valve.

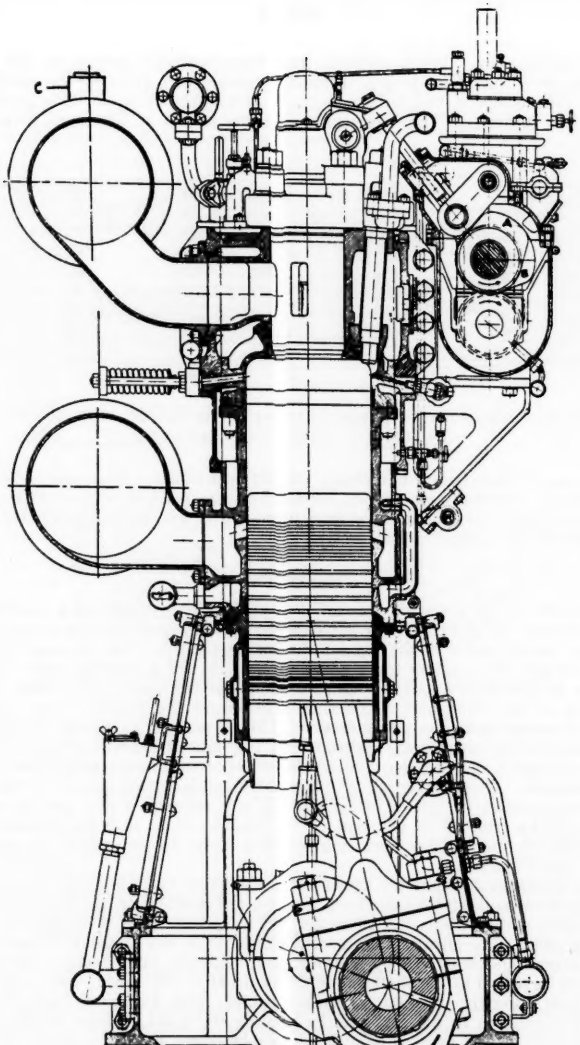


FIGURE 7.—SECTION THROUGH THE COMBUSTION CHAMBER OF A GENERATOR.

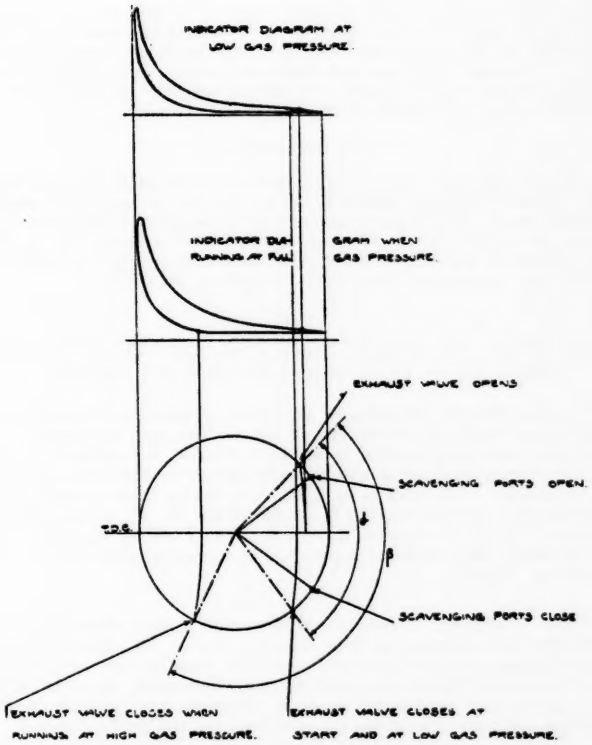


FIGURE 8.—VALVE DIAGRAM.

When maneuvering, the generators may not stop when the gas delivery occasionally ceases. The generators are therefore provided with an escape valve for exhausting the gas mixture through a pipe to the open air. This valve is operated by a governor on the generator which opens the valve when the number of revolutions of the generator comes down to about one-quarter of the normal RPM. This prevents the generator from stopping.

It is proposed that in certain cases the generators be of the opposed-piston type and with the compressor pistons attached to the upper combustion piston. This is, for instance, the case with the generators shown in the proposal, Figure 6.

The heat drop on the expansion of the gas is comparatively low in relation to the heat drop for steam expanding from a high pressure to a good vacuum. If gas at a pressure of 5 kg./cm.^{2} absolute and a temperature of 425 degrees C. is allowed to expand to atmospheric pressure, the heat drop is 62 kcal./kg. gas. When steam at a pressure of 30 kg./cm.^{2} and a temperature of 350 degrees C. is allowed to expand to a vacuum of 0.05 kg./cm.^{2}

absolute, the heat drop is 252 kcal./kg. steam. The heat drop on expansion of gas is thus only about one-quarter of the heat drop on expansion of steam. On account of this, only a few rows of turbine vanes are required for converting the energy of the gas into mechanical work. For a marine turbine the ahead turbine usually requires one impulse and four reaction wheels and the astern turbine one impulse wheel only.

PIPE DIMENSIONS.

As the gas pressure is low the pipes will attain comparatively large dimensions. These dimensions, however, do not become so large that practical difficulties arise. With a gas velocity of 50 m./sec. in the inlet pipes to the turbines, the area of the pipes becomes 0.3 cm.^{2}/BHP or 0.0465 sq. in./BHP. If the velocity of the exhaust gas is determined at 75 m./sec., the area of the exhaust pipes becomes 0.67 cm.^{2}/BHP or 0.104 sq. in./BHP.

FIELD TESTS OF FREQUENCY- AND AMPLITUDE-MODULATION WITH ULTRAHIGH-FREQUENCY WAVES.

This paper presents comparative field tests of ultrahigh-frequency transmitters using amplitude-modulation as against frequency-modulation. The results are of particular interest in presaging a major development along new lines. Frequency modulation will be advantageous in voice transmission at high frequencies. The paper is by I. R. Weir, Radio Transmitter Engineering Department, General Electric Co., Schenectady, N. Y. It was printed in the General Electric Review of May, 1939. Part I only is reproduced here. Part II, which deals with the application of frequency modulation to radio networks, is omitted.

The use of frequency modulation offers certain distinct advantages in the transmission and reception of radio signals. During the past few years a considerable amount of analytical and laboratory work has been done with this type of modulation *†‡ and some field tests have been made. It is the purpose of this article first to describe a series of recent field tests at ultrahigh frequencies and then to point out the significance of the results obtained.

ANTENNAS AND LOCATIONS.

A 50-watt frequency-modulated transmitter and a 50-watt amplitude-modulated transmitter were located in Schenectady. A 150-watt frequency- and amplitude-modulated transmitter was located 15.5 miles away, in Albany. The antenna at each location was the vertical "J" type. The Schenectady antenna was 465 feet above sea level or about 240 feet off the surrounding ground level. The Albany antenna was located 610 feet above sea level or about 390 feet above the surrounding ground.

Measurements were taken at intermediate points between these transmitters with a frequency- and amplitude-modulation receiver mounted in a test car. The receiving antenna consisted of a short, nongrounded car antenna of the vertical-rod type.

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* "A Method of Reducing Disturbances in Radio Signalling by a System of Frequency Modulation," by Edwin H. Armstrong, Proceedings I. R. E., vol. 24, p. 689, May, 1936.

† "Frequency Modulation Noise Characteristics," by Murray G. Crosby, Proceedings I. R. E., vol. 25, p. 472, April, 1937.

‡ "Noise in Frequency Modulation," by H. Roder, Electronics, vol. 10, p. 22, May, 1937.

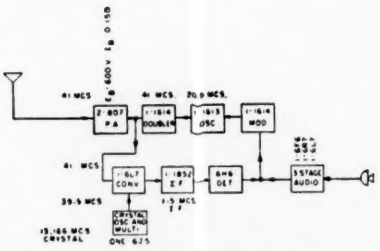


Fig. 1. Block diagram of a 50-watt frequency-modulated transmitter

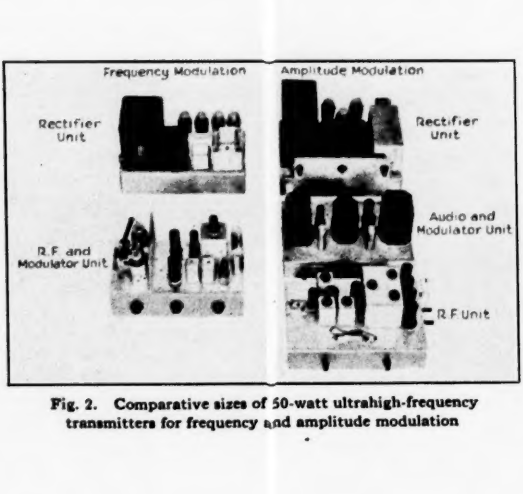


Fig. 2. Comparative sizes of 50-watt ultrahigh-frequency transmitters for frequency and amplitude modulation

TRANSMITTERS.

The frequency-modulated transmitter circuit used in these tests maintains a high degree of stability by the use of automatic frequency control. This same means is used for frequency-modulating the carrier at an audio-frequency rate. It is in this way that a simple frequency-modulated transmitter with a high degree of stability is obtained without the necessity of many frequency-multiplication stages.

Each of the two Schenectady transmitters had the same carrier output power of 50 watts. The amplitude-modulated transmitter was of a conventional plate-modulated type. The frequency-modulated transmitter is shown in Figure 1, which indicates the tubes and frequencies employed. The transmitter proper consists of a master oscillator tube driving a doubler tube which in turn excites two amplifier tubes connected in parallel.

Since the oscillator of this circuit must be frequency modulated (low-level modulated), it is necessary to have some method to hold the mean frequency of the modulated oscillator at the fixed carrier frequency of 20.5 megacycles. This stabilization is accomplished by feeding the 41-megacycle output of the doubler tube into a converter tube where it is heterodyned to an intermediate frequency by means of a 39.5-megacycle crystal oscillator having very high stability. This intermediate frequency (1.5 megacycles) is passed through a frequency-discriminating circuit, then amplified and rectified (detected) so that the rectified energy may be used to vary an element voltage of a reactance tube connected across the tuned circuit of the modulated oscillator. Thus, the mean frequency variations of the oscillator are converted to energy which changes a reactance in the oscillator circuit in a manner to compensate for the mean frequency variation tendencies.

This frequency stabilization system is slow acting, so that signalling frequencies also may be applied to an element voltage of the reactance tube which will modulate the frequency of the oscillator, but which will not be compensated by the automatic frequency-control circuits. A conventional audio amplifier system is fed into the grid of the modulator tube as shown in Figure 1. This varies the modulator bias at an audio-frequency rate around the normal bias established by the frequency stabilization circuit.

A physical comparison between the frequency- and amplitude-modulated transmitters is shown in Figure 2. The frequency-modulated transmitter is about two-thirds the size of the amplitude-modulated transmitter. The total weight of the former is 85 pounds, while the amplitude-modulated transmitter weighs 105 pounds. The total input power taken by the frequency- and amplitude-modulated transmitters while delivering equal carriers is 325 and 625 watts, respectively. The frequency-modulated transmitter does not require an increased power input with 100 per cent modulation.

A schematic circuit of the 150-watt transmitter used at Albany is shown in Figure 3. Provision was made whereby regular amplitude modulation could be removed and excitation for the last two amplifier stages obtained from a 50-watt frequency-modulated transmitter.

MEASUREMENT EQUIPMENT AND TEST PROCEDURE.

The receiver used in taking all amplitude- and frequency-modulation comparison measurements described in this article consisted of a double-conversion superheterodyne. Provision was made to switch the receiver from diode detection for amplitude modulation to frequency-modulation detection by using a limiter and a special detector circuit which has been described * by Major Armstrong. Figure 4 is a block diagram of the test receivers.

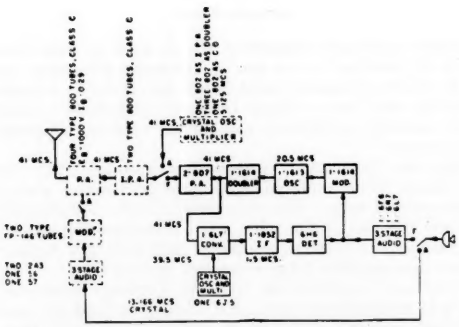


Fig. 3. Block diagram of a 150-watt transmitter, designed for either frequency modulation (switch position *F*) or amplitude modulation (switch position *A*)

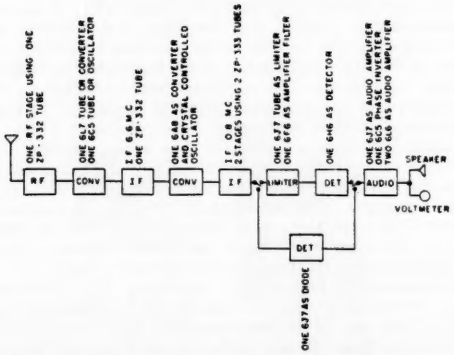


Fig. 4. Block diagram of receiver. Switches up for frequency modulation; switches down for amplitude modulation

The selectivity curve (Figure 5) was 220 kilocycles wide at twice down (6 db) and 530 kilocycles wide at one thousand times down (60 db). The selectivity was made this wide at twice down so that it would be possible to receive a ±80-kilocycle frequency-modulated signal and take care of the transmitter and receiver stability.

The audio characteristics of the receiver measured ±2 db of the 400-cycle response from 60 to 10,000 cycles.

One of the characteristics measured during these tests was signal-plus-noise to noise ratio. This quantity was measured by a two-scale copper-oxide voltmeter connected across the output of the receiver. The ratio of the audio output voltage of the receiver when receiving a modulated signal to the audio output voltage when receiving an unmodulated signal of the same carrier strength is the "signal-plus-noise to noise ratio." This ratio should not be

confused with the input of the receiver or "r-f signal-plus-noise to noise ratio."

A d-c milliammeter was connected to the automatic volume control circuit of the receiver in order to measure the receiver input voltage. A signal generator was used to calibrate the automatic volume control meter.

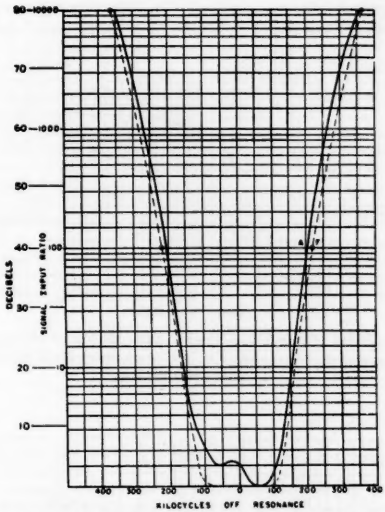


Fig. 5. Selectivity (Curve *A*) of a 15-tube receiver with amplitude modulation and (Curve *F*) values obtained from two frequency-modulated transmitters on separate channels. (Receiver output equal for each transmitter.)

In order to compare the frequency- and amplitude-modulated systems under identical conditions, all possible elements were kept the same with the two systems of operation. The transmitting and receiving antennas, the transmitter output stages, and transmitter carrier power all remained unchanged. The same receiver was used, the only change consisting in the use of different types of detectors.

Field Tests.

SIGNAL-PLUS-NOISE TO NOISE RATIO.

Field measurements were first made with signals from the Albany transmitter. A constant plate voltage and plate current were maintained on the output stage for both frequency and amplitude modulation.

Figure 6 shows the ratio of signal-plus-noise to noise versus field strength taken in areas free from external interference. These areas were found some distance from power lines and other sources of interference. The flattening off of the frequency-modulation ratio above 10 microvolts field is due to the

fact that the audio amplifiers in the transmitter at Albany were not perfectly free of noise. It will be noted that there is a decided difference between frequency- and amplitude-modulation in favor of frequency modulation.

Figure 7 indicates the same measurements as shown in Figure 6 but made in areas with fairly high external noise level. The external noise was about the level one might find in a downtown business district. These curves show a marked difference in the two types of modulation. Frequency modulation shows a maximum improvement of about 25 db with equal carriers.

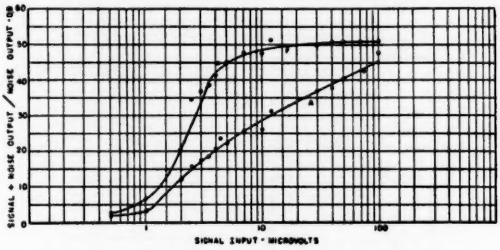


Fig. 6. Ratio of signal-plus-noise to noise output of (*F*) frequency-modulated receiver and (*A*) amplitude-modulated receiver without external noise

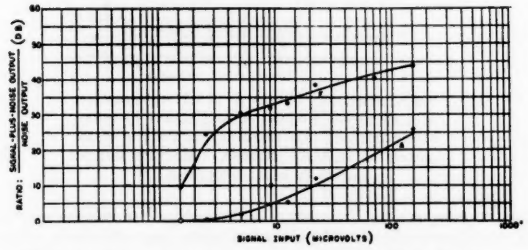


Fig. 7. Ratio of signal-plus-noise to noise output of (*F*) frequency modulated receiver and (*A*) amplitude-modulated receiver with external noise

The automobile which carried the frequency-amplitude test receiver was parked in several quiet areas and signal-plus-noise to noise measurements were taken with the ignition suppression equipment removed from the engine. During these measurements the back wheels were jacked up so the engine could run at different speeds, creating a variable interference level while the desired signal remained constant. The results of this test are plotted in Figure 8. These measurements show that amplitude modulation is more affected by interference. A signal-plus-noise to noise ratio of 4 to 1 or 12 db requires a field strength of 2 microvolts for amplitude modulation and only 1½ microvolts for frequency modulation with no external noise. When the automobile was

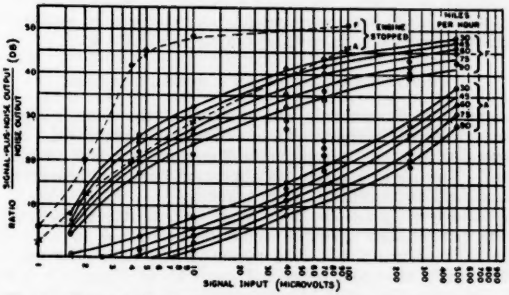


Fig. 8. Ratio of signal-plus-noise to noise output of (*F*) frequency-modulated receiver and (*A*) amplitude-modulated receiver with car-ignition noise

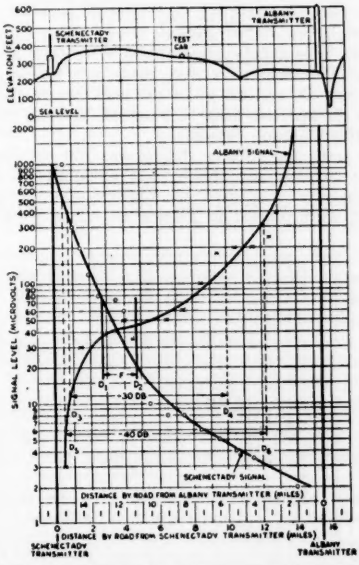


Fig. 9. Signal levels of two frequency-modulated transmitters at Albany and Schenectady. With frequency modulation, both signals could be heard intermittently in the area *F* between D_1 and D_2; with amplitude modulation, interference areas are shown between D_3 and D_4, D_5 and D_6

"running" 90 miles an hour, amplitude modulation required a receiver input of 70 microvolts while frequency modulation required a receiver input of approximately 3 microvolts to give an output ratio of 4 to 1.

TWO-STATION OPERATION ON A SINGLE CHANNEL.

The transmitter in Schenectady was operated on the same frequency as the Albany transmitter (41 megacycles) and frequency modulated. A different program was fed to each transmitter so the stations could be identified by the program. The radio test car was operated between these two stations, Figure 9, and field measurements were taken.

Only the Albany program could be heard out to 10.8 miles from Albany and only the Schenectady program could be heard out to 2.7 miles from Schenectady. There was a two-mile region, where both programs could be heard intermittently as shown between D_1 and D_2 in Figure 9. Very rarely could both programs be heard simultaneously in this two-mile area. The movement of the car a few inches would throw one program out and bring in the other. This effect was due to a standing wave pattern caused by local reflections which were superimposed on the average field-strength curve of Figure 9. *Wherever the field strength of one signal was greater than twice the other field strength, the stronger signal would prevail 100 per cent.* This region should not be considered an interference area as would occur with amplitude modulation, but rather an area where either signal may be heard.

Both Albany and Schenectady transmitters were then amplitude-modulated on 41 megacycles. The same tests were made as with frequency modulation. The distance between D_3 and D_4 gives the area where the interference was greater than —30 db and the area between D_5 and D_6 indicates where the interference was greater than —40 db.

After these two tests, the Albany station was 100 per cent modulated with 700 cycles and the Schenectady station was modulated with 400 cycles. A radio-frequency analyzer was used to measure the receiver output. The results of this test were plotted in Figure 10. This shows that, if the de-

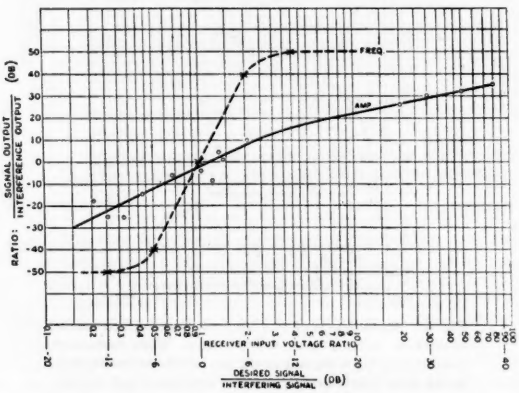


Fig. 10. Frequency- and amplitude-modulation interference between two signals of approximately the same frequency (41 mc)

sired frequency-modulated signal is greater than twice the undesired signal, the desired signal takes complete control. The desired amplitude-modulated signal must be at least thirty times the undesired signal before the interference is low enough for suitable broadcast reception.

TWO-STATION OPERATION ON ADJACENT CHANNELS.

The Albany transmitter was operated on 41 megacycles and the Schenectady transmitter was operated on 41.5 megacycles. Both transmitters were frequency-modulated with different programs. The receiver in the test car was tuned to 41 megacycles. The signal characteristics between the two transmitters was investigated and at no point was the 41.5-megacycle signal heard even when the test car was parked directly under the 41.5-megacycle antenna.

Then the Schenectady transmitter was operated on 41.25 megacycles while the Albany transmitter remained on 41 megacycles. The receiver was tuned to 41.0 megacycles. No interference was detected from the 41.25-megacycle signal until the test car was within one-half mile of the 41.25-megacycle transmitter.

It was found that 360 kilocycles was the minimum carrier frequency separation possible to operate the receiver tuned to the Albany transmitter under the Schenectady antenna without any interference from Schenectady. The field-strength ratio between the desired and undesired signal was greater than 10,000. The dotted curve in Figure 5 indicates the ratio measurement taken of equal signals from the receiver as compared to the receiver-selectivity curve (heavy line). The point to bear in mind while comparing the amplitude-selectivity curve with the frequency-modulation curve is that a variation of a few kilocycles will completely eliminate the undesired frequency-modulated signal while with amplitude modulation it would give a strong interfering signal.

ADVANTAGES.

The tests described in this article thus reveal several advantages of frequency modulation over amplitude modulation which may be summarized as follows:

(1). Improved signal-plus-noise to noise ratio. Under some conditions this improvement is as high as 20 to 25 db. This means there is a remarkable freedom from atmospherics and man-made static such as X-rays, automobile- and aircraft-engine ignition, commutator sparking, etc.

(2). A more definite and uniform service area of a transmitter is established because frequency modulation signal-plus-noise to noise ratio remains high until the field intensity reaches a very low value.

(3) Comparatively much smaller geographical interference area is obtained when two frequency-modulated transmitters are operated simultaneously on the same frequency than when amplitude-modulated transmitters are so operated.

(4). A frequency-modulated radio-frequency amplifier is more efficient than one for amplitude modulation because frequency modulation can be accomplished at low level followed by class "C" power amplification.

(5). A given service area can be covered with considerably less power than with amplitude modulation because of the improvements in signal-plus-noise to noise ratio obtained with frequency modulation.

(6). For a given power output, smaller radio-frequency amplifier tubes can be used.

In considering these advantages, it should be particularly noted that the design and operation of transmitter equipment for frequency modulation is

no more complicated than for amplitude modulation of equivalent power. The design used in the present tests was smaller and much lighter in weight. The frequency-modulated receiver is about equal in size and weight to an amplitude-modulated receiver and employs about the same number of tubes.

RESEARCH IN THE PROBLEMS OF FLIGHT.

George W. Lewis, F.I.Ae.S., delivered the Twenty-seventh Wilbur Wright Memorial Lecture to the Royal Aeronautical Society on May 25, 1939. He traced the course of aerodynamic research from the early wind tunnel of the Wright brothers to the more advanced developments of today. The abridged lecture is extracted from the June 9, 1939, issue of Engineering, London, England.

The brothers Orville and Wilbur Wright made a remarkable contribution to aeronautics by flying the first successful power-driven aircraft, of which both the engine and airframe had been constructed by them from the results of their experimental research. Through the courtesy of Orville Wright, the author is able to give an account, hitherto unpublished, of the experimental work that had contributed to the success of their efforts.

The original tunnel of the Wright brothers was 5 feet long, had a square test section of 22-inch sides, and could produce an air velocity of 40 feet per second. Their original research program covered a comprehensive investigation into the behavior of circular arc plates with systematically varied camber. The models had a 1-inch chord and a 6-inch span, giving an aspect ratio of six, which was now the standard for aerofoil tests in most aerodynamic laboratories. Extensive investigations of the effect of plan form, aspect ratio, biplane gap-chord ratio, and other factors were also conducted. This was indeed a surprising research program for the year 1901, and so clear was the Wright brothers' appreciation of the basic factors in the problem that they included all but one of the important variables that had since concerned us in wing and aerofoil research.

In the original apparatus employed by Orville and Wilbur Wright for the measurement of fundamental forces, two balances were employed, one for measuring lift directly and the other for evaluating the ratio of drift/lift as the tangent of an observed angle. The lift was balanced directly against a set of resistance plates, which had previously been calibrated against the drag of a square flat plate of known area, in the same position in the air stream as the wings to be investigated. The drift, or drag, was measured in per cent of the lift. Thus far the steps taken were mainly to obviate difficulties resulting from non-uniform and non-constant velocity in the air stream, although care had previously been taken to reduce these to a minimum. Still, however, no direct measurement of actual forces to be expected on the full-size wings was available. This was accomplished by the use of a glider tested in 1902. This glider had a wing scaled up from one of the models tested in the wind tunnel, and by means of glide tests in winds of known velocity, it was possible to determine the lift and drag as actual forces. These tests, in effect, calibrated one of the wings tested in the wind tunnel for full-scale conditions, and since the relation of the other wings to this wing was known from the wind-tunnel measurements, the characteristics to be expected from all the wings, when flown full size, were directly indicated by the wind-tunnel data. This description from Orville Wright revealed one aspect of the Wright Brothers' research work that had never been realized before—the fact that they adopted an expedient in the year 1901 that auto-

matically corrected measurements at small Reynolds number to the Reynolds number encountered in flight. By the steps described they reduced the critical assumptions to one: that the order of merit of the aerofoils would not be seriously affected by Reynolds number, though the absolute values of the coefficients might change. It was not until about 30 years later that aerodynamic research settled this question and showed that, at least for a number of critical aerodynamic characteristics, the assumption was valid. One other point of interest in this connection deserves note. Using the average flight measurements on their 1902 glider and the wind-tunnel results on models, the Wright brothers calculated that the unit drag of a square flat plate of large area was equal to the square of the speed in miles per hour multiplied by the coefficient 0.0033. At the time, they had their choice among figures varying from 0.0025 to 0.0055, based on the tables of other investigators. Much subsequent research had led to general adoption of the figure 0.00328 for the cofficient in question; truly astonishing agreement with the value reached by the Wright brothers through averaging a series of results obtained in flight measurements on a glider in the year 1902.

Following the establishment of large Government laboratories for the study of aerodynamic problems, scarcely a year had elapsed in which new contributions to wind-tunnel design had failed to appear. The general trend of improvement had been in increased size of the wind tunnel to permit the use of larger models; increased speed of the air stream to keep up with the increase in performance of the aeroplane; and provision for investigations at larger Reynolds numbers by the use of high-density air. The improved design of wind tunnels had been dictated largely by the necessity to solve the problems that arose in aeroplane design. With designers' vision frequently reaching far in advance of contemporaneous development, it had been necessary to improve the wind tunnels year by year, so that data obtained from model tests could be directly applied to the design of new types of larger and faster aeroplanes. Since the days of its early growth, the National Advisory Committee for Aeronautics had endeavored to emphasize this aspect of aeronautical research, and had therefore followed the policy, not only of undertaking theoretical analyses of aeronautical problems, but, in so far as possible, of anticipating the future needs of aeroplane design by constructing wind tunnels having special characteristics. This policy was illustrated by the following equipment at Langley Field:—The variable-density wind tunnel for aerofoil research at high Reynolds numbers, the 20-foot propeller-research tunnel, the full-scale tunnel, and the high-speed jet tunnels for aerofoil research under propeller operating conditions. These tunnels, and the results of their work, are well known in aerodynamic circles.

More recently, anticipation of further increases in aeroplane performance led to the construction of the committee's 8-foot 500-MPH wind tunnel. The results of investigations in this tunnel had proved effective in providing design information for the development of high-speed aeroplanes. It might be said that the design of new wind tunnels of such types as this was predicated almost entirely on present and future problems that were apparent from a survey of the performance characteristics anticipated for future military and commercial aircraft. Realization of the expected advances in aircraft development, as to size, as well as speed, had made it necessary to augment fundamental or basic research with applied research on special models. Such models, complete with control and power plants, and having spans up to 40 feet, were now investigated in the full-scale wind tunnel. The careful investigation of highly refined models was well repaid when these represented large aeroplanes such as had recently been built or contemplated. The Committee's recent efforts in this policy were exemplified in the new 19-foot pressure tunnel, shown in Figure 6. This tunnel, designed in 1937, was now

being placed in operation. Having a throat diameter of 19 feet, it could accommodate large-scale models to be tested at pressures from sub-atmospheric up to several atmospheres, and at air speeds in excess of 250 MPH. It was planned to use this tunnel in extending the types of investigation that had been carried on in the propeller-research tunnel and in the full-scale tunnel. A marked trend during the past few years had been towards the use of dynamic models for investigation of the stability and control characteristics of aeroplanes. The initiation of this useful method was due to the Aeronautical Research Committee in their laboratories at Farnborough, where the original free-spinning wind tunnel was developed. Others had followed, and at present every major aeronautical research laboratory had included a free-spinning wind tunnel as part of its equipment.

The first, and probably most important, aerodynamic problem faced by the aeroplane designer was that of the speed, range, and efficiency of the aeroplane. These factors could all be considered under the broad field of aerodynamic efficiency, in which the chief objective was the reduction of drag. Investigations in aerodynamic laboratories had mainly centered around the characteristics of the wing, even as did those of the Wright Brothers, because it was simple and represented many of the fundamental problems of aerodynamics. The classical theory of wings of finite span further simplified the problem, so that the principal effort could be devoted to investigation of the characteristics of aerofoil sections. Turbulence and velocity fluctuation in the air stream of wind tunnels had long been a matter of concern in the broad field of fluid mechanics, as well as in aerofoil research proper. For the interpretation of wind-tunnel results to conditions existing in the free atmosphere, a helpful simplification resulted from the concept that turbulence in the atmosphere was of such a nature that its effect on the transition from laminar to turbulent flow in the boundary layer of an aeroplane wing might often vanish under free-flight conditions. Data tending to confirm the validity of this concept were obtained independently in Germany and in the United States about four years ago.

Values of the critical Reynolds number of transition in excess of 6,500,000, had been found when interpreted in terms of the equivalent flat plate Reynolds number used by B. M. Jones. The maximum values of the equivalent flat plate Reynolds number were those given by Jones as approaching 5,000,000. Comparisons made to date suggested that the transition was hastened in flight by other disturbing effects. In the tunnel, disturbances such as surface roughness and vibration were carefully avoided. Nevertheless, it still did not appear probable that the turbulence had been reduced to the effective zero level, but the level appeared sufficiently low to justify certain conclusions based on the tunnel investigations, pending further research in flight.

In many ways the preliminary results of these investigations had proved illuminating. Under conditions of vanishing turbulence, it appeared that transition might be of a different character in a low-turbulence tunnel than in the usual tunnel. The laminar boundary layers ahead of transition often followed accurately the laminar boundary-layer theory, and appeared to be free, or nearly free, from unsteadiness, or fluctuations of the type observed by Dryden. Thus the skin-friction drag produced by these laminar layers at the comparatively large Reynolds numbers attainable, was no greater than that predicted by the laminar boundary-layer theory. Aerofoils theoretically designed to take advantage of true low drag laminar boundary layers over a major portion of their surfaces, show spectacular results from the standpoint of drag reduction when tested under suitable conditions in a two-dimensional flow tunnel. These statements might best be illustrated by Figure 8, showing a bank of manometer tubes connected to a wake-survey apparatus so as to measure the total head effect behind an aerofoil mounted

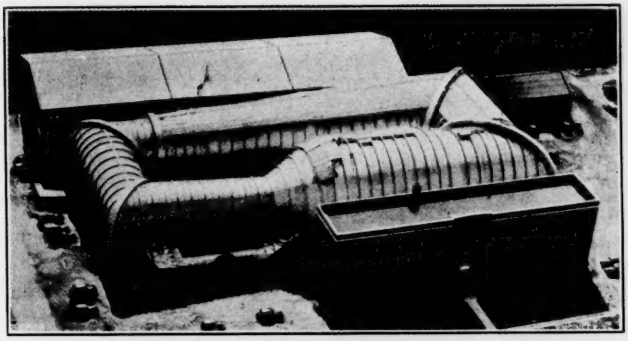


FIGURE 6.—19-FT. PRESSURE TUNNEL.

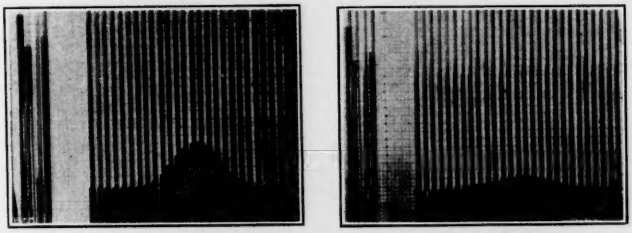


FIGURE 8.—COMPARISON OF WAKE BEHIND AEROFOILS.

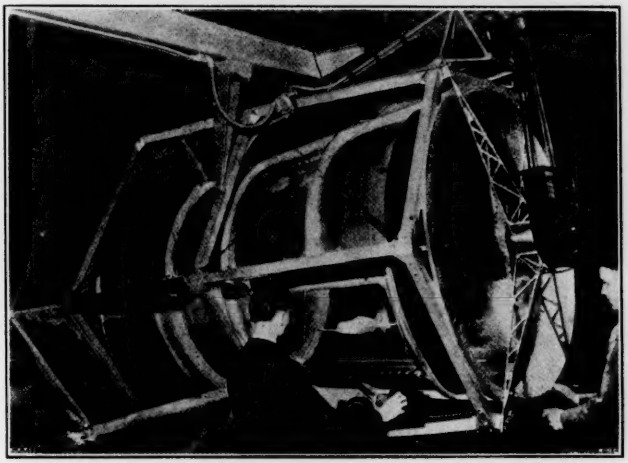


FIGURE 16.—5-FT. FREE-FLIGHT TUNNEL IN OPERATION.

in the low-turbulence tunnel. The illustration on the left showed the wake behind a smooth polished model of an N.A.C.A. 0012 aerofoil at zero lift. The illustration on the right was for one of the new aerofoils, in this case at a lift coefficient of 0.2. In other respects the test conditions were identical, the Reynolds number being approximately 5,000,000. It was evident from these illustrations that, in the lower Reynolds number range, the new aerofoils yielded drag coefficients of a different order from those heretofore obtained. Work now in progress included attempts to extend the laminar boundary layers to still higher values of the Reynolds numbers, as well as other phases of the subject.

The structural integrity of aircraft, as affected by aerodynamic loads, had been the major concern of a group on the Committee's staff under the supervision of Richard V. Rhode for a number of years. Messrs. Lundquist and Donely, as well as other members of the Committee's staff, had contributed to the results thus far obtained. To state, then, the problem of aerodynamic loads, it had been evident for many years that atmospheric gusts gave rise to loads of considerable magnitude on aeroplane structures. With the continuing increases in the size and speed of transport and bombing types, it had become increasingly apparent that these gust loads constituted the most important conditions for which the structure should be designed, because (1) the relative structural weight tended to increase with increasing size, making it necessary to reduce the design loads of large aeroplanes to a minimum; (2) increased size was, in general, accompanied by sufficiently increased wing loading to result in greater gust loads, on account of the reduced alleviation caused by the motion of the aeroplane with the gust; and (3) the gust loads increased directly with the speed of flight. In anticipation of the coming importance of the problem of gust loads, the available data on the structure and intensity of atmospheric gusts, and on the accelerations experienced by aeroplanes flying through them, were reviewed in 1930. As a result of this survey it seemed desirable to obtain extensive acceleration data under actual operating conditions, in order that a sound basis for the establishment of gust-load criteria might be provided. It was further apparent that air-speed measurements should be made simultaneously with the acceleration measurements, in order that the results might be reduced to a reasonably rational common denominator, viz., an "effective" gust velocity. Since the cooperation of pilots was desired, it was necessary to devise an instrument so simple and reliable that there could be no reasonable objection to installations on transport aircraft. For this and other practical reasons, an instrument, now known as the N.A.C.A. V-G recorder, was designed. This instrument combined an accelerometer element and an air-speed element, in such a way that the stylus plotted simultaneous values of the two quantities on a smoked-glass plate during any desired period of operation.

This type of instrument had made possible, with very few units, the collection of the desired extensive data. There had been accumulated on commercial aeroplanes up to March 20, 1939, records covering 67,300 hours of flying time, representing approximately 10 million miles under widely varied conditions of topography, weather, etc. With the records, it had been possible to ascertain that the probable maximum effective gust velocity (based on the sharp-edge gust hypothesis) was about 30 feet per second, for an aeroplane having the characteristics of the Boeing 247, and this value was currently used as a basis for the determination of gust-load factors in the United States. In order to obtain experimental data on gusts, a "gust tunnel" had been developed* which allowed a model to perform a short period of free flight under the influence of artificial gusts. The flight path

* Donely, Phillip. "An Experimental Investigation of the Normal Acceleration of an Aeroplane Model in a Gust." T.N. 706. N.A.C.A. 1939.

of the model was recorded on a cinematograph film from which the velocity could be computed. This tunnel had provided useful data for the designer, and in one case it had been found safe to select gust-load increments, 17 per cent less for the light condition and 22 per cent less for the heavy condition, than the values that would have been chosen without the benefit of the tunnel.

Problems of stability and control had long been recognized as one of the major factors in the aerodynamic design of an aeroplane affecting safety in flight. In this field, in particular, a survey of the literature revealed very extensive activity on the part of all major aerodynamic laboratories. Yet with the development of newer, larger and faster aeroplanes of novel design, both broad and detailed problems continued to arise, and the field remained as active as ever. In the early stages of the general stability and control investigation, it rapidly became evident that the problems of stability and control were intimately associated with the development of high-lift devices demanded by the design trend towards higher wing loadings. Further, it was evident that the most pressing problems then related to lateral stability and control, principally at high angles of attack. On this basis, extensive investigations were made in the 7-foot by 10-foot wind tunnel, covering all promising suggestions for improved high-lift and lateral-control devices that came to the attention of the Committee. These investigations were supplemented in the full-scale tunnel and in flight, on such arrangements as were found promising in the smaller wind tunnel. Several developments arising from the investigation were now coming to be accepted by a number of designers for certain classes of aircraft. Among these might be included the reduction of stalling hazards by the limitation of longitudinal control and by the use of novel landing techniques.

Recent design trends had brought the question of the detailed stalling characteristics of three-dimensional wings more critically to our attention, and it was now recognized that these constituted a design problem in themselves, distinct from the general stability and control characteristics in the normal flight range. The wing-tip slot, originally developed in England by Mr. Handley Page and his associates, was recognized as an effective expedient for improving lateral stability and control near the stall. It had been found that the dynamic longitudinal stability of an aeroplane, as calculated from the classical theory, had little relation to the pilot's judgment of its flying and handling characteristics. This, though a negative result, suggested a field of investigation in refinement of the aeroplane's responses, on the one hand to accidental disturbances encountered in flight, and on the other hand to the characteristics and capabilities of the human being as a pilot.

Another, perhaps more practical, field of investigation was concerned with the characteristics of new aeroplane designs. Particularly in the case of large and expensive aeroplanes it appeared that a new method of predetermining the flying and handling characteristics was desirable. On this basis, and with background experience in development of a free-spinning tunnel, following the introduction of the type at Farnborough, the Committee felt that the attempt to develop a free-flight tunnel was amply justified. Figure 16 showed the present 5-foot diameter tunnel in operation. The tunnel itself had only one unconventional feature. It was suspended on trunnions so that the inclination could be varied from horizontal to about 25 degrees below the horizontal. Air speeds up to 25 feet per second could be attained. As shown in the illustration, two operators were required. The operator beside the tunnel had the task of maintaining the inclination and air speed in proper co-ordination with the gliding characteristics of the model under test. The operator of the model at the end of the tunnel had several functions to perform; firstly,

he must maintain the model laterally in the center of the tunnel by use of the rudder to compensate for lack of stability in azimuth; secondly, he must operate the controls on the model in studies of unsteady flight conditions; and thirdly, he must observe the general flying and handling characteristics of the model. This last function followed almost automatically from the difficulty encountered in achieving the desired flight conditions. Operation of the controls was effected by the use of small electromagnets mounted in the nose of the model. The electromagnets were supplied with current through a fine trailing wire, of sufficient length to remain slack, connected to the model-operator's switches.

The procedure in a test run was somewhat as follows: With the model on the floor of the tunnel the angle was adjusted to about the expected value, and the air speed was slowly increased. When flying speed was attained the model took off and climbed, the speed and angle then being adjusted to attain steady flight in the center of the tunnel. Meanwhile, the "pilot" had the sole task of maintaining the model laterally in the center of the tunnel by use of the rudder. He then took charge of the flight conditions and proceeded with the test schedule. No extensive investigations had yet been undertaken in this tunnel, although numerous model flights had been made in the course of the development work. To investigate the utility of the tunnel for direct study of the stability characteristics of an aeroplane, a model was fairly extensively tested. This model represented an existing areoplane which was known to have certain unsatisfactory characteristics in its original form. Owing to the low scale of the tests, the investigation was largely confined to conditions away from the stall, although it was found possible to simulate the stalling characteristics of the aeroplane roughly by empirical adjustments of the filleting. The first step was to consider the possible effect of longitudinal stability of the handling characteristics of the aeroplane, by comparing the behavior of the model with normal horizontal tail area and with the area increased 50 per cent. Briefly, it was concluded that the longitudinal-stability characteristics did not appreciably affect the problem in this case. The remainder of the investigation was, therefore, directed to the effects of various combinations of dihedral and fin area on the flying qualities of the model.

Although no quantitative verification in flight had been undertaken, observations on the full-scale machine appeared to substantiate this conclusion. A more satisfactory aspect of these results was the excellent qualitative agreement between the trend as to ease of handling found in the tunnel and the trend to be expected from theoretical considerations already well known. On this basis, the free-flight tunnel appeared to offer a useful, rapid and relatively inexpensive means for the investigation and correction of the handling characteristics of new aeroplane designs. As a result of the work on the 5-foot tunnel, the Committee was sufficiently encouraged to undertake the construction of a larger free-flight wind tunnel having a throat diameter of 12 feet. This tunnel, contained in a large spherical steel shell, could be operated at a pressure of several atmospheres, and would provide a means for extending the model research on stability and control to larger sizes and higher Reynolds numbers.

In the course of this lecture, I have endeavored to show how the National Advisory Committee for Aeronautics, through the application of organized research on the problems of flight, has attempted to meet the needs of aeroplane design by the development of specialized research equipment. Where the Wright Brothers led, many men indeed have followed, and this new branch of human activity, still less than forty years old, has reached sufficient proportions to justify the prediction that it will never come to an end so long as man is on the earth. Such progress as can be reported has been the result

of much individual and collective effort, and in this conclusion, I wish to acknowledge the sincere and whole-hearted co-operation of the whole staff at Langley Field for their contributions to the N. A. C. A.'s share in this development.

Finally, I sincerely hope that the work they have been able to carry on in the United States will prove of interest to their friends and co-workers in the field of aeronautics in England.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

ELECTRICALLY-HEATED ZINC BATHS FOR HOT GALVANIZING.—Extract from The Brown Boveri Review, Baden, Switzerland, April, 1938.

Hot galvanizing has always played an important part as a protection of iron parts subjected to the vagaries of weather or a damp atmosphere. After preliminary cleaning in an acid solution, the iron parts to be thus protected are dipped in a hot fluid zinc bath, which causes a coating of zinc of about 0.1 millimeter thickness to adhere to the iron. When, afterwards, the zinc coating is subjected to damp air or to rain, a coating of zinc oxide carbonates forms on the surface, which prevents further oxidization and, thus, rusting of the iron part.

The ordinary commercial kinds of pure zinc are extracted by electrolytic processes from zinc ores, such, for example, as zinc blend (ZnS), calamine (galmey) (CO_3Zn), etc., and have a zinc content of up to 99.9 per cent. The melting point of pure zinc is 420 degrees C. For hot galvanizing, the zinc bath is heated up to 450 with a maximum of 460 degrees C. By immersion of short duration in the liquid bath, the iron gets covered with a coating of the desired thickness. As is well known, iron has a very great affinity to zinc, that is to say tendency to solubility and this is much enhanced at high temperatures. Therefore, in order to carry out good hot galvanizing and to assure economical operation of the galvanizing plant, it is very important that the working temperatures of the zinc baths be maintained at exactly the desired temperature.

The melting tanks themselves are made of pure soft iron or, in some cases, of Armco iron of low carbon content. The melting tank is gradually worn away by the hot zinc and even though the wall thickness may be 30 to 50 millimeters, the tank must be changed after a certain time, depending on the heating method applied and the operating conditions of the plant.

In the majority of galvanizing works, coal or oil firing is still in use. Apart from the relatively poor thermal efficiency of these firing systems, it is a difficult problem to attain uniform heating throughout the melting tank. In order to prevent the heating flames, with temperatures up to 1300 degrees C., from coming into direct contact with the external walls of the tank, the said walls are protected by a layer of fire clay. Despite this, it is impossible to prevent certain regions of the melting tank—and this especially during stoppages of operation—from suffering from excessive heating. Now, as is well known, the solubility of overheated surfaces is increased and the formation of a spongy iron-zinc mass or hard zinc, is accelerated, which is translated into a quicker wearing out of the tank. It may, even, happen that the tank bursts at some exposed spot, so that the fluid zinc runs out into the furnace. An

accident of this nature not only means expensive repairs, but a putting out of service of the plant affected for a considerable period.

Thorough scientific investigations showed that the rapidity of the hard zinc formation is, mainly, dependent on the temperature. The great importance of the temperature is shown by the fact that material recognized as suitable for a zinc melting bath has a 3.2 times greater solubility at 500 degrees C. and a 12.5 times greater one at 550 degrees C. than it has at 460 degrees C. It is, therefore, understandable that zinc baths heated by fuel must be very closely supervised and every effort made to prevent high temperatures appearing at any spot in the bath proper or melting tank.

Apart from the accelerated destruction of the tank and from the loss of zinc due to the formation of hard zinc, there are other reasons why the plant supervisor should supervise the temperature of the zinc bath, closely. When the temperature of the bath is too low, the coating of zinc on the pieces treated is irregular and too thick; therefore, too much zinc is used up, uselessly. Further, the adherence of the zinc layer is poor and it flakes off easily when the piece galvanized is worked up or if the galvanized object is used. In the case of too high temperatures, the layer is rough and unpleasing. Further, the surface of the tank oxidizes at too high bath temperatures, which is, again, the cause of loss of zinc.

The *electric heating* of zinc melting tanks offers some very great advantages; thus, it is possible to heat up the external walls of the tank absolutely uniformly, by a suitable arrangement of the resistances and the whole external surface of the tank is utilized for heat transmission, at minimum temperatures. Automatic temperature regulation prevents the admissible temperature limit from being exceeded. In this way, the desired bath temperature is maintained exactly and automatically, independently of the production passed through the bath, and the objects being galvanized are covered by a uniform, thin zinc coating. In one case of a big galvanizing plant, with electric heating, better zinc consumption and considerably less hard zinc formation was noted, when electric heating was adopted, within a short time. It is obvious that the melting tank itself will have a longer life due to less wear, thanks to the lower and evenly-distributed temperatures at all points; this has a most favorable influence on the costs of the plant.

For many years, now, Brown Boveri have been carrying out the electrification of galvanizing plants, and this with the greatest success.

An electrically-heated galvanizing plant was built in 1931 and used for the galvanizing of iron wires. With a power input of 30 Kw., a production of up to 1000 kilograms in 24 hours is recorded, here. Exact measurements showed that this fine galvanizing worked out at a power consumption per kilogram of material passed through the bath of 0.4 Kwh., for a wire diameter of 1.3 millimeters and of 2 Kwh. for 0.17 millimeter wire diameter. The consumption of zinc referred to the weight of the wire showed an average of 18 per cent. The melting tank is equipped for automatic temperature regulation and the furnace calls for no supervision at all and produces continuous, uniformly-distributed galvanization of high grade.

An electrically-heated galvanizing plant was also built for the hot galvanizing of big containers, pipes, girders, etc., by full immersion. The melting tank has a capacity of 40,000 kilograms of melted zinc. The maximum power input of the furnace is 180 Kw. Supply is by three-phase current at 8000 volts, through a transformer with taps corresponding to various load steps. Thus, the heating resistances in the furnace can be connected up to different secondary voltages according to the heat requirements and these vary between 130 and 380 volts. In the case of big parts, such as girders or pipes, work is carried out with open bath surface. A big part of the heat input is carried

away through radiation from the open surface of the bath. For work of this kind, with heavy galvanized parts, the plant works on correspondingly higher output steps on the supply transformer. In the case of smaller receptacles, sheet-metal wares, etc., a smaller bath surface suffices so that the part of tank not being utilized is covered in with heat-insulating covers and the heating resistances are supplied at a lower voltage. In continuous service the plant produces from 7 to 10 tons of iron in 24 hours at a power consumption of about 0.2 Kwh. per kilogram of galvanized product.

The heating resistances are arranged on all four walls of the melting tank in such a way that there is perfectly uniform radiation from the said resistances and uniform heat transmission to the melted zinc of the bath. The heating power is distributed between separate regulating groups governed by automatic switching contactors. The furnace temperatures are measured by means of thermo-elements which indicate the temperature to the temperature regulators. The desired temperature of the bath can be adjusted for on the temperature regulator, very simply. If the heating power brought to the bath is excessive, the corresponding switching contactors are switched out and the output of the corresponding heating group is automatically reduced. Thus, overheating of the bath itself or of the tank at any point is quite impossible. The lower zinc consumption and slighter formation of hard zinc which results from these favorable conditions was perceived very soon.

The brickwork of the furnace is composed of fire-clay bricks of great heat resistance, which carry the heating elements. In order to avoid heat losses to the foundations, a very effective heat insulation made of special material is placed behind the fire-clay covering. During stoppages in operation, the whole bath is covered in with heat-insulating covers, in order to maintain the heat stored in the bath as long as possible.

As will be clear from the foregoing paragraphs, the melting tank is gradually worn away, due to the solubility of iron in hot zinc and it is, often, difficult for the service operatives to know when the proper moment has come to put in a new melting tank. It may happen that material faults in the tank are the cause of slight leaks appearing after a short period of service which then enlarge rapidly so that there is a considerable flow of melted zinc into the furnace brickwork. In order to detect possible small leaks in the tank, gutters are provided in the floor of the melting-tank furnace. Two contacts are located at the lowest point of these gutters, which actuate an alarm signal as soon as they are brought into contact with liquid metal. In a case of this kind, the bath can be emptied immediately in order to prevent further and worse damage.

The running costs of the electric melting bath are lower than those of fuel-fired plants, at the present metering rates, so that, quite apart from the greater cleanliness and simplicity of electric supply, it is advantageous to change over from fuel firing to electric heating. The electric furnace works with a uniform load and a power factor of 1 and it is, therefore, a welcome type of consumer to the power-supply companies, this the more so as most galvanizing plants operate continuously night and day.

In order to make comparative calculations on plant economy, it would not be correct to calculate the metering cost of current only with relation to the pure heat equivalent of the fuel-fired plant. As has been said, the consumption of zinc in the melting tank can be considerably reduced by using electric heating; further, there is a lower percentage of hard zinc and slag of zinc oxide. Further, the running costs are favorably affected by the elimination of salaries to stokers and, last, but not least, by the longer life of the melting tank and the elimination of costly repairs. The proper numeral evaluation of all these advantages is not, however, an easy matter and may have varying results, according to the plant under consideration.

The practical results reached with electrically-heated galvanizing plants have given ample confirmation in all points of the advantages expected when the plants were projected, so that galvanizing plants, desirous of keeping up to date when getting new equipment or when carrying out big repairs, find it to their advantage to go over to electric heating.

ICE BREAKER BLADES REQUIRE TOUGH STEEL.—Nickel Steel Topics, the International Nickel Company, New York, N. Y., April, 1939.

The Canadian Car and Foundry Company, Ltd., recently completed an order for eight new propeller blades to be installed on the ice breaking boats used by the Canadian Government for keeping harbors open during the winter months. This is a repeat order, and since the original use of this material for this purpose several years ago, it has been found that no other cast material is so well suited to resisting the combination of impact and wear encountered in this service.

It is reported that breakage, which was at one time fairly frequent, has been eliminated by using this steel in the normalized and drawn condition. Average composition and properties follow:

Carbon, per cent	0.20
Manganese, per cent	0.90
Silicon, per cent	0.50
Nickel, per cent	1.95
Vanadium, per cent	0.12
Yield Point, p.s.i.	67,500
Ultimate Tensile Strength, p.s.i.	93,000
Elongation in 2 inches, per cent	30
Reduction of Area, per cent	55

Bend Test—160 degrees—No fracture.

SOME FACTORS IN THE LUBRICATION OF HIGH-SPEED DIESEL ENGINES.—Mechanical Engineering, New York, N. Y., May, 1939. (Extract from a paper by A. T. McDonald, Caterpillar Tractor Company, Peoria, Ill., presented at a regular meeting of the Central Illinois Section of The American Society of Mechanical Engineers, at Peoria, Ill., September 15, 1938.)

Satisfactory lubrication of an internal-combustion engine, once taken for granted as being accomplished when the crankcase was filled with oil, has developed into one of the most complex and involved problems for the modern engine builder. This is due, in part, to the rapid advances made in engineering and design in the last few years, which have resulted in greater pressures, higher speeds, and higher outputs of engines. This is particularly so in the case of aircraft and high-speed Diesel engines.

The rapid development of the high-speed Diesel engine and its acceptance as a dependable and economical source of power brought about the realization that the demands placed upon ordinary lubricants by this type of engine were far beyond their limits of stability, oiliness, and film strength. Because high-speed Diesel engines are most satisfactorily lubricated by special lubricants, and, since this paper deals primarily with the lubrication of this type of engine,

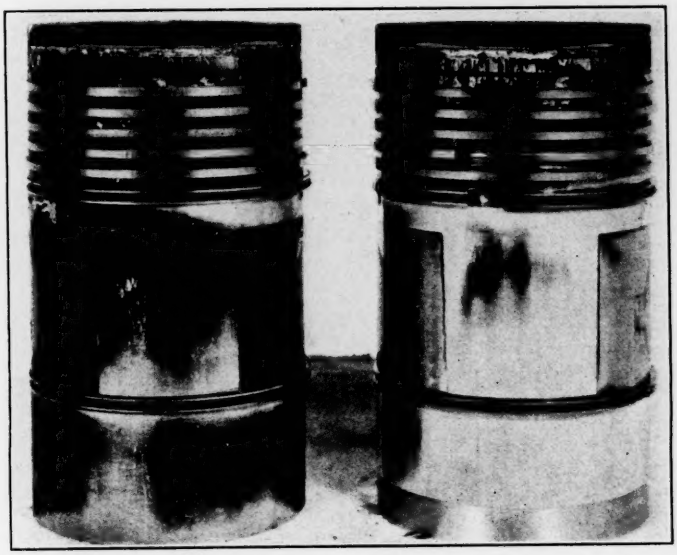
the author feels that it is advisable at this point to give a brief review of the development of specially manufactured lubricants for this purpose.

The arrival of efficient high-speed Diesel engines, with their higher pressures and more critical piston-ring belt temperatures resulted in frequent failure of the engine due to liner scuffing or to piston-ring sticking. After exhaustive experimental activity in the way of engineering design had failed fully to justify these failures, serious thought was given to a study of the lubricant in seeking a solution to the problem.

The first step along these lines was the further treatment of lubricants to achieve a state of higher chemical stability (as measured by various accelerated oxidation tests). This, in most cases, resulted in an increase of difficulties rather than a decrease. Cooperative research by the engine builders and the lubricant manufacturers disclosed the fact that overtreatment of lubricating oils was not only possible, but also as injurious to the lubricating qualities of a lubricant as insufficient treatment. After exhaustive tests, certain stocks were selected as representing the optimum in balance between purification and the retention of all desirable lubricating qualities. Still, even these carefully manufactured lubricants fell far short of providing satisfactory lubrication. It was at this point that additive agents were introduced to increase film strength, oiliness, and resistance to oxidation. These compounds, when added to basic stocks of proper origin and treatment, proved of inestimable value in the lubrication of high-speed Diesel engines.

An example of the relative merits of straight mineral oil and a specially compounded lubricant designed for the lubrication of a high-speed Diesel engine is shown in Figure 1 in which *(a)* represents a piston which had operated 375 hours in a Caterpillar Diesel engine using as a lubricant a straight mineral oil, and *(b)* represents a piston after 2000 hours' operation in the same engine under identical operating conditions using the same oil plus one per cent of an additive agent commonly used in the manufacture of Diesel-engine lubricants. The effectiveness of this type of additive is quite apparent and needs no further explanation here. There has been so much misconception, however, of the manner in which these additives function, that for purposes of clarification, the following description of this phenomenon seems to be in order.

Failure through ring sticking is generally brought about in two ways. First, the chemical decomposition of the lubricant results in the formation of a hard, lustrous, and practically insoluble lacquer-like material on the piston surfaces, which, combined with carbonaceous material, acts to cement the ring firmly in its groove. Second, lack of film strength in a lubricant results in metal-to-metal contact of the bearing surfaces in the high-pressure regions, causing rings and liners to scuff or score, which results in abnormal blowby with consequent poor combustion and causes sooty or gummy depositions, or both, from combustion which, when combined with excessive depositions from the lubricant caused by high temperature as a result of the high blowby, also cement the ring in its groove. In the first case the additive acts partially as an oxidation inhibitor, reducing the quantity of deleterious materials formed. In addition to this, it has the effect of increasing the detergent quality of the oil. This added detergency or cleansing power has the effect of removing, during the incipient stages of their formation, the aforementioned carbonaceous and binder materials. Consequently, bearing surfaces stay relatively clean and the rings remain free. The second and equally important function of the additive is to increase film strength to such an extent that sufficient safety factor is provided to prevent metal-to-metal contact under the most severe conditions of pressure and temperature.



(a) *(b)*

FIGURE 1.—TWO PISTONS FROM SAME ENGINE.

[*(a)* Operated for 375 Hours With Mineral Oil; *(b)* Operated for 2000 Hours With Mineral Oil Containing an Additive Agent.]

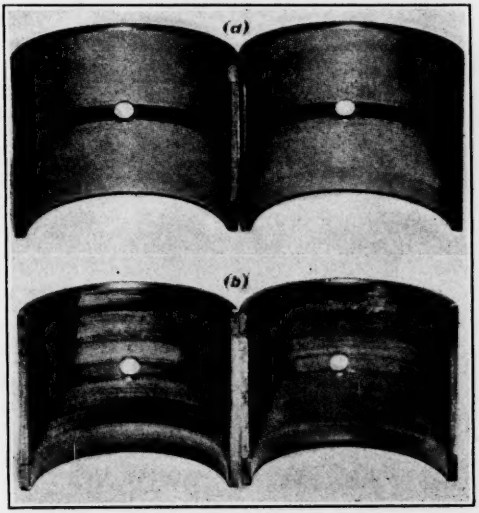


FIGURE 3.—EFFECT OF CHANGING AND NOT CHANGING OIL ON BEARINGS.

[*(a)* A Bearing After 2500 Hours of Operation With Oil Changed Every 60 Hours, and *(b)* Another Bearing Run in the Same Engine for 500 Hours Without Changing Oil.]

All lubricating oils contain, in varying percentages, natural oxidation inhibitors, and film strength and oiliness compounds. The additive agents used in the manufacture of special Diesel lubricants (metal soaps) are added, not to give an oil new qualities, but to increase the effectiveness of those already inherent in the lubricant.

Another very important problem has been that of bearing lubrication. Bearing failures are caused in most cases by excessive temperature. Pressures with respect to bearing failure are only of secondary importance. This can be readily appreciated if the structural change which takes place in an average babbitt bearing with increase in temperature is considered. The ultimate strength of an average babbitt bearing at a temperature of 77 degrees F. is approximately 16,000 pounds, at 200 degrees F. the ultimate strength drops to 9000 pounds, and at 279 degrees F. it has been reduced to less than 1200 pounds. It is evident then that temperature, not pressure, is the pertinent factor to consider when investigating bearing failures.

There are many causes for the development of excessive temperature in connecting rod and main bearings. One common and little considered cause is that of the introduction of foreign material, such as silica, small wear particles, or metal filings, into the lubricating system. This material is carried by the oil stream to the bearings and gradually becomes imbedded in the bearing metal, changing a precisely manufactured bearing into an excellent lapping tool. A bearing in this condition gradually wears the journal out-of-round. It has been found that this type of contamination causes wear rates as high as 0.001 inch per 100 hours of operation. A shaft worn to an out-of-round condition touches the bearing unevenly, causing local hot spots, eventually weakening the babbitt structure to an extent where even nominal pressures will result in failure. A third type of bearing failure is that of poor bonding of the bearing material to the shell proper. This is a type of failure that was once quite common, but one that has been almost eliminated. Research by bearing manufacturers has developed methods for bonding bearing materials that have apparently solved this phase of the bearing problem. Another type of failure, and one that unfortunately is not uncommon, is that of bearing corrosion. The formation of corrosive oxidation products with the chemical decomposition of a crankcase oil sometimes results in a bearing becoming etched to a degree that will cause failure. The etching process brings about the formation of microscopic craters and cracks in the bearing surface. These become filled with minute droplets of oil, and when pressure is applied a hydraulic action is set up, forcing the oil deeper into the bearing surface and eventually breaking away sections of the bearing. The direct cause of this type of failure with normal lubricating oils is again high temperature, which is proved by the fact that practically all types of crankcase lubricants and their products of decomposition at temperatures under 200 degrees F. under normal service conditions are in themselves noncorrosive. Table 1 illustrates this conclusion.

Another little considered cause in the development of high temperatures that eventually result in bearing failure is the use of crankcase lubricants for extended periods beyond their limits of useful life. This practice results in increased viscosity or excessive thickening of the lubricant. A reduction in oil flow with consequent lessening of heat dissipation through this source can, and does, result in structural change in the bearing material which is conducive to bearing failure. Figure 3 is an illustration of this type of bearing failure wherein *(a)* represents a bearing taken from an engine which has run 2500 hours with the lubricant changed every sixty hours. From the appearance of these bearing shells, there can be no doubt that many hundreds of hours of satisfactory service still remain in them. In *(b)* are shown bearing

TABLE 1.—COMPARATIVE CORROSION OF BABBITT METAL AND CADMIUM SILVER AS A FUNCTION OF TEMPERATURE.

	Cadmium Silver Weight Loss, Milligrams.			
Temp. F.	*Oil A*	*Oil B*	*Oil C*	*Oil D*
209	0.0	0.0	0.2	0.0
240	0.2	37.0	1.3	7.2
270	0.6	50.3	3.7	9.6
300	0.6	27.7	22.0	23.1
330	0.6	30.3	21.0	31.7
360	2.2	9.9	20.6	12.4
	Babbit Metal Weight Loss, Milligrams.			
209	0.0	0.0	0.3	0.3
240	0.8	0.3	2.1	0.6
270	0.5	0.6	10.5	0.9
300	1.9	17.1	68.5	7.7
330	5.0	26.8	72.6	8.5
360	17.9	71.8	107.3	26.6

shells which had run in the same engine under identical operating conditions with respect to load, etc., except that the oil was not changed for a period of 500 operating hours. In this test every effort was made to keep foreign or carbonaceous material from being introduced or left in the system by means of effective filtration of the lubricant. In spite of these precautions, it is apparent that this bearing has already failed. In other words, the bearing problem seems to consist of the control of two main factors. First, the control of crankcase and operating temperature to a point where structural change of the bearing metal is minimized and the corrosive tendency of crankcase oils reduced to assure a sufficient safety factor; and second, the elimination, so far as possible, of the imbedding of foreign material in the bearing.

In this day and age of controversial opinion, no paper on this subject would be complete without some reference to crankcase-oil filtration or purification. The use of effective methods of filtration is of the utmost importance and, without question, has a definite place in the scheme of lubrication. It is important to point out here that the main function of a lubricating-oil filter is the removal of solid contamination from the lubricating oil. The difficulties encountered in the use of filters have been largely due to misleading advertising by various filter manufacturers. Exorbitant claims in the past have led to the use of this type of equipment as a cure-all for every conceivable trouble ever encountered in an engine or with a lubricating oil. Considering that not over ten per cent of the lubricating oil used has any lubricating value, the remaining 90 per cent acting only as a coolant and as a vehicle for "lubricity" compounds; that chemical decomposition takes place in some degree throughout the whole lubrication cycle; and that the compounds doing the work are most affected by such reactions; then the fallacy of attempting to use lubricating oils for extended periods, over and above that recommended by the engine builder who knows his engine and by the lubricant manufacturer who knows his product, is easily recognized. A filter should be considered as a means of removing solids from an oil without any beneficial refining influence.

Adsorbent or chemically active filters when used with compounded oils remove the additive agent almost immediately and are not generally accepted as being of particular value.

Too much emphasis cannot be placed upon the fact that filters are used to increase the life and efficiency of an engine, not as a means of increasing the life of the lubricant for, as described in the foregoing, lubricating-oil life is

controlled by chemical reactions; the degree and severity of the reactions, governed by engine design and operation, are solely responsible for oil deterioration. The removal of contaminants, whether they result from chemical decomposition of the oil, or from outside sources, does not retard these reactions.

In conclusion, it can be said that in spite of the amazing progress made in recent years in the field of lubrication, this field of endeavor is still in its infancy. Much remains to be done by the engine builder and by the lubricant manufacturer. It is hoped that further cooperative effort will result in developments that will lead to the elimination of many present-day problems.

OUT OF THE MIRE: ALSIFILM.—The Industrial Bulletin of Arthur D. Little, Inc., Cambridge, Mass., March, 1939.

Dr. Ernst A. Hauser reports that a new type of transparent film material can be made from clay. He suggests his "Alsifilm" as a mica substitute, an indestructible paper, a fireproof wrapping material and a waterproof coating for paper articles.

The clay used is an especially "pure" type of bentonite, made up of alumina, silica and water, and the individual particles are so small that most of them fall within the colloidal range. It has been found that the finer part of the colloidal fraction of the clay when mixed with the appropriate amounts of water forms a gel. If this gel is evaporated, the particles draw toward one another in such positions that they become permanently fixed by their attraction for each other, in strings or tiny fibrils. These strings in turn mat together similarly to the matting of wood fibers in forming paper, or wool in felt, so that ultimately the resulting structure is that of a tough, coherent, independent film.

Alsifilm may be made thick so that it is stiff and brittle, and in this state is proposed as a substitute for mica, for it looks like mica, has similar electrical properties, and is chemically inert. Both mica and Alsifilm are composed of Alumina, silica and bound water, possibly in much the same proportions and distribution.

In presenting his findings to the Technical Association of the Pulp and Paper Industry last month, Dr. Hauser stated that the new film may be made completely inert to oils, greases, organic solvents and strong acids and salts. There are obvious useful applications for such a material available as a film or a thread, and possessing these properties. In the thin, flexible film form, for cable-wrapping, it would offer long life and the needed electrical properties. Foods, especially those which tend to take up odors and tastes readily, might be packaged in this inorganic transparent film. Dr. Hauser also indicated that Alsifilm will take ordinary printing, colored inks and printing pastes, or it may be colored throughout, filled and coated, oiled, or given a surface pattern. Unaffected by climate, mold, or insects, it is seen as a permanent record material, handled as easily as any paper.

Alsifilm is to be introduced commercially by the Research Corporation for the Massachusetts Institute of Technology and Dr. Hauser. At the present stage of development none of the material is available for distribution, but a small commercial unit is being constructed and samples will be available within a few months. The cost of Alsifilm cannot well be estimated until further progress toward the production of larger quantities has been made.

DISCUSSIONS AND COMMENTS.

INVESTIGATION OF THE ACTION OF CONDENSER SCOOPS BASED UPON MODEL TESTS.

(SEE PAGE 191, MAY, 1939, JOURNAL.)

HENRY F. SCHMIDT, WESTINGHOUSE ELECTRIC AND MANUFACTURING COMPANY.—Professor Weske's article is based entirely on the hypothetical assumption of no frictional, eddy or shock losses and, furthermore, assumes that it is permissible to put the necessary water through a pump or condenser at an enormously high velocity in order to conserve this velocity so that it does not constitute a loss. Unfortunately, in the design of Naval and Merchant propelling machinery it is necessary to deal with real water and real materials which are not free from frictional resistances and other sources of energy loss.

As the question of the energy distribution in the wake belt and what effect it has on the actual losses in taking circulating water on board and discharging it, is the most controversial point in deciding the design of the circulating water system, the following mathematical proof that the energy loss in picking up water from any point in the wake belt is exactly the same as though it were taken from still water is of fundamental importance and will be of interest to all Naval engineers.

The following is a simple mathematical treatment, prepared by Mr. R. P. Kroon,* of the case in which fluid is transferred from the boundary layer into a ship by means of a scoop.

For this analysis, the wall of the ship has been taken to be straight and it has been assumed that the disturbed flow around the scoop covers only a small portion of the length of the ship.

In Figure 1, there is shown the velocity profile of A-A in front of the scoop. The flow into the scoop (which is taken to extend

* Division Engineer, Westinghouse Electric and Manufacturing Company.

along the whole depth of the wall) disturbs the flow, but let us suppose that at B-B, a certain distance behind the scoop, there has been reestablished a boundary layer with a velocity profile exactly equal to that at A-A.

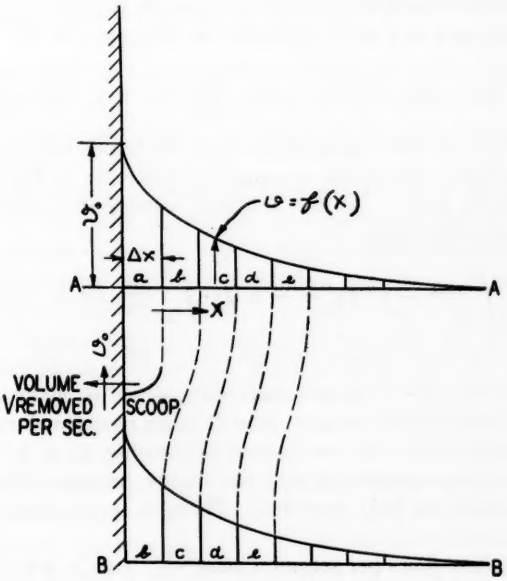


FIGURE 1.

Since fluid is taken out from a region which has already obtained the maximum absolute velocity, other portions have to be accelerated.

A simple conception of this can be obtained by dividing A-A into a number of elements a,b,c,d, etc. of equal width Δx. One can then imagine that the fluid going through element a is entirely taken out by the scoop. It is imagined to be replaced by element b of A-A, which now becomes the boundary element b at B-B. Similarly, the elements c,d,e, . . . of A-A are supposed to find corresponding places at B-B. (It should be pointed out that as

long as we are interested only in the total amount of fluid supplied and the total energy involved, it does not matter whether the dotted line in the figure are physical streamlines or not.)

The above conception facilitates a mathematical formulation. It is seen that any particle which had a velocity $v = f(x)$ at A-A is accelerated to a velocity $v^* = f(x - \Delta x)$ at B-B.

Taking Δx as a small quantity, we may put:

$$v^* = f(x - \Delta x) = f(x) - \Delta x \cdot \frac{df}{dx} \ldots\ldots = v - \Delta x \cdot \frac{dv}{dx} \ldots\ldots$$

The volume V taken out by the scoop in one second can immediately be calculated, since it has to equal the difference in flow through the elements b,c,d . . . at A-A and B-B, respectively. With h being the depth of the fluid,

$$V = h \int_{\Delta x}^{\infty} (v^* - v)\, dx = - h\, \Delta x \int_{\Delta x}^{\infty} \frac{dv}{dx}\, dx = - h\, \Delta x.$$

$$v_{x=\infty} + h\, \Delta x \cdot v_x = \Delta x$$

Since v at $x = \infty$ is zero and since, when we make Δx small enough, the velocity at $x = \Delta x$ will come nearer and nearer v_o, the volume taken out per second approaches $V = h\, \Delta x \cdot v_o$. Similarly, it is possible to find the energy necessary for the acceleration of the fluid particles. Through every element Δx at A — A there flows per second a mass $\frac{\delta}{g} h\, \Delta x\, v$ which carries in an energy $\frac{\delta}{g} h\, \Delta x \cdot v^3$. At B — B the corresponding element passes an energy $\frac{\delta}{g} h \cdot \Delta x \cdot v^{*3}$.

Therefore, the energy put in per second by the surrounding fluid is

$$E = \frac{\delta}{g} h \int_{\Delta x}^{\infty} (v^{*3} - v^3)\, dx.$$

Since $v^{*3} = v^3 - 3 \cdot \Delta x \cdot v^2$ ..... this can be written

$$E = \frac{\delta}{g} h \int_{\Delta x}^{\infty} - 3 \cdot \Delta x \cdot v^2 \cdot \frac{dv}{dx} \cdot dx$$

Since $\int v^2 \cdot \frac{dv}{dx} \cdot dx = \frac{v^3}{3}$ this becomes:

$$E = \frac{\delta}{g} h \Delta x \left[v^3_{x=\infty} - v^3_{x=\Delta x} \right]$$

approaching $\frac{\delta}{g} \cdot V \cdot v_o^3$ for the limit $\Delta x = o$

It is seen that the energy increase (which, of course, has to be supplied by the ship) is equal exactly to the energy one has to exert to accelerate the volume of water removed from zero velocity to the full ship velocity.

The hydrodynamical relationships which Professor Weske has derived are, of course, correct for the conditions assumed, but the writer feels that because of the difficulty of distinguishing between purely hypothetical mathematical relations and the actual case, this paper may do considerable harm. The impression which the paper may give many readers is that it is only necessary to take one of the intakes shown in the paper, install a pump, and, since for the ideal fluid with no losses all the energy in the velocity of approach is regained at the discharge, that a pump is a far more efficient piece of apparatus than a scoop.

Regarding the experimental results the writer would severely criticize the method of procedure if these tests were actually carried out in a closed passage of a wind tunnel as indicated by the text. It is extremely difficult to measure the static pressure with great accuracy in a moving stream in a closed passage. The author does not give the dimensions of the closed passage in which the test apparatus is installed, but it is basically incorrect, in making measurements in a wind tunnel, to inject or remove fluid from the passage in which the measurements are being taken. In the case of these experiments a layer of air was scooped off in order to vary the boundary layer thickness and also through the scoops and out of the overboard discharges tested. For this reason serious doubt is cast on the accuracy of the test, since the maximum velocity employed corresponded to approximately only 2 inches of water, so that errors of only a few tenths of an inch in measuring the static pressure of the moving column would cause a

serious error. This is further indicated very clearly by the test curves given in Figures 4, 5 and 6. These are for an intake scoop and show that the total head decreased with increase of scoop capacity.

Without any tests, ordinary experience has taught us that a very short length of straight pipe has relatively no frictional losses and we approach the condition of a wide open piece of pipe as the capacity of the intake is increased. Under this condition the velocity through the pipe would nearly equal the velocity of approach, that is, its efficiency as expressed by e_t should approach unity. The tests, however, show a decrease in efficiency.

The writer tested a common scoop with lip similar to those tested by the author, results of which were presented in the February, 1930, issue of the A. S. N. E.* The results to the same system of coordinates used by the present author are shown in Figures 38 and 39. Figure 38 is for the same condition as the test under discussion.

These tests, it will be observed, conform with theoretical expectations, that is, the total head increases with the capacity approaching to near unity, whereas Professor Weske gets about 41 per cent for the same percentage capacity. Also, the static head or c_p should not reach zero until substantially 100 per cent normal capacity is reached. That is, a ratio of $\frac{v_1}{v_0}$ of unity.

It is impossible to discuss the performance of the overboard discharges, since there are obvious errors in the test of the scoops and there are no similar means of checking the accuracy of the overboard discharges, which, however, undoubtedly suffer from the same causes as the tests of the scoops.

While Professor Weske appears to have primarily written the article to assist designers of pump systems to improve the efficiency of their arrangements, all the overboard discharge data given is for discharges without lips, which necessitates designing the pump for a higher head.

Because of the disturbance of the wake belt by the overboard discharge, the deflection of the wake belt to make room for the

* See Figure 38, page 29, and Figure 39, page 30, February, 1930. JOURNAL.

water discharged is performed for much less power when accomplished by a lip than by the circulating water.

A properly designed lip in effect, produces a groove into which the discharge flows with a minimum of mixing and turbulence in the surrounding water.

For this type of work, where it is necessary to inject or remove quantities of fluid from the moving stream, the jet should either be of extremely large cross section, compared with the model, or else the jet should be in the open, as in the writer's test. This applies to test with water as well as with air, and the fact that it has been attempted to test model scoops in a propeller test tunnel undoubtedly has led to errors for the same reason. A jet discharged under water with a free surface obviates these errors. An open gap type wind tunnel such as the 50-foot tunnel at Langley field would be satisfactory.

In regard to the pressure recovery from the 90-degree overboard discharge, this can be explained very simply by the fact that since the water is deflected rearwards by the relative velocity of the water passing the ship, this causes the water to flatten out against the hull. This results in a contraction of the stream of water leaving the overboard discharge, thus producing a reduction of pressure at the small radius (the after side of the scoop pipe) which induces a forward thrust and is also responsible for the measured back pressure in the overboard discharge when a lip is not employed.

However, while it is true that in the actual scoop there is a recovery from a 90-degree discharge, this recovery is relatively small and is very much outweighed by the disturbances to the wake belt and the consequent increase of frictional resistance aft of the discharge.

The method of recording the results of the tests is not suitable for application to actual problems, since the performance is based on the undisturbed velocity past the ship, whereas the wake belt from ¼ to 1 inch thick was very large compared to the diameter of the test scoops. Because of the way in which the wake belt was varied, the distribution in the wake belt is unknown or at least not given, and probably in any event does not approximate the relationship given by von Karman's formula, whereas the wake

belt around a ship does so very closely. In any event, to be applicable for practical purposes the performance of the scoop should be given in terms of the integrated value of the velocity of approach instead of the jet velocity.

Referring to the discussion of the results, the discrepancy between the measured and calculated values in the inlet scoop is attributed to separation along the up stream wall of the scoop and the formation of a pair of vortices at the forward edge of the intake. It is suggested that the scoop can be improved by increasing its width or "aspect ratio."

Just the reverse is true, since from an efficiency standpoint it is desirable to get the highest velocity of approach possible and by making the scoop narrower and taking water from farther out the advantage gained by the higher kinetic energy of the available water far outweighs the slight losses in these vortices. This was clearly indicated by the writer's experiments. The writer tried scoops with trapezoidal inlets. One set of tests was run with the wide part of the scoop at the forward edge of the entrance and another test with the widest part at the after edge of the intake. Both these forms of intakes were inferior to the intake with a rectangular opening, resulting from making the normal section of the scoop at the inlet a square.

The only logical reason for making the whole inlet of a scoop wider is in order to enable a design to be made for high speeds with a sufficiently short radius to make it compact and light without danger of cavitation.

The writer agrees with paragraph B—the discussion of results—in that if sufficiently high air velocities are used in experimental work, no scale effect can be detected between the model in air and the full-size scoop in water.

Under the paragraph headed Effect of Lip, it is suggested that the maximum efficiency for intake scoops for use with a pump occurs at a ratio of $\frac{v_1}{v_0} = .6$. As will be pointed out later, from an energy standpoint, the ideal would be attained could a scoop which offered no resistance be designed which would reach out from the

ship into undisturbed water so as to utilize the full velocity of approach and with an inlet area just sufficient to admit the desired quantity of water at the velocity of approach and reduce this velocity and reconvert the energy in a divergent tube to the velocity which is permissible through the eye of the pump runner. In this case the actual efficiency could be brought quite close to unity.

Referring to the author's Figure 3—test set-up—there is nothing to indicate that any guide vanes were provided above the propeller to prevent rotation of the air in the upper portion of the pipe. The pressure tap for obtaining the static pressure from which the flow through the orifice plate was obtained is quite close to the plate. The pressure recorded would be seriously affected by rotation produced by the propeller.

The characteristics of pumps and scoops are absolutely identical, the pressure varying as the square of the volume, the speed remaining constant. In the case of the pump, however, while the efficiency remains constant for an equivalent constant orifice, the efficiency of the system at reduced capacities such as required at cruising speeds if a pump is used alone, becomes extremely low, due to the inefficiency of the driving means, whether turbine or motor. It has been proposed to use pumps of the movable blade type, but these are also very inefficient at the very low per cent of capacity required at cruising speeds and the prime movers also are very low in efficiency because of the very light load—ten per cent or less.

As previously pointed out, a pump designed to handle the very high velocity required to avoid the waste of kinetic energy in the intake with common scoops would be very inefficient, so that divergent inlets which reduce the velocity of the water to that suitable for the pump inlet are a necessity in any case. If, however, the velocity is reduced to a point which is suitable for admission to a pump, the conversion of velocity into pressure in the scoop is such that the pump is not needed.

The writer would take issue with the introduction to Professor Weske's article; first, in the statement "Beyond this, the several investigators of past years have not demonstrated in practice a superior method of scoop design."

When the divergent type of scoop was first proposed, there was considerable worry about the ability of a scoop with considerably

less than ⅓ the area of the then existing scoops, to actually deliver more water to the condenser than had previously been obtained, but it was always pointed out that scoops gave excessive water at reduced powers, resulting in very cold condensate.

The writer at that time proposed a divergent scoop with a movable inlet lip which would be capable of being automatically or manually adjusted to give the required volume of water at low speeds. Because of the fear, however, that the divergent scoop would not work in any event, the idea of further reducing the inlet area was not favorably received. This was in the latter part of 1929, but, as it has not previously been described, the following notes may be of interest.

Since the impact pressure recorded by an impact tube projecting from the ship's hull indicates a pressure which is directly proportional to the square of the ship's speed, if this is impressed upon a piston resisted by a spring, as shown in Figure 2, it is obvious

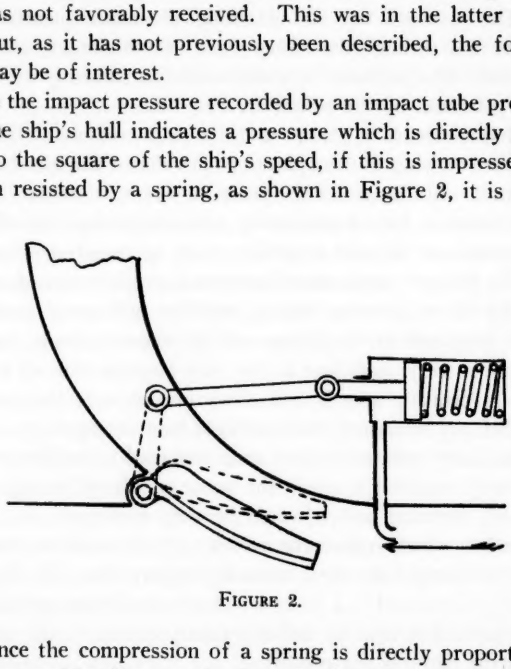


FIGURE 2.

that, since the compression of a spring is directly proportional to the force, and in this case the force is directly proportional to the square of the static pressure, if this piston is connected to the movable flap as indicated, the flap will be caused to move so that the area of the inlet of the scoop varies exactly as the square of the ship's speed. Also, since the velocity of approach to the scoop is directly proportional to the velocity of the ship and the area of the

inlet varies as the square of the speed of the ship, we see that the quantity of water entering the scoop will be directly proportional to the cube of the ship's speed as is also the horsepower. Consequently, roughly, the ratio of water to steam will remain substantially constant. However, the power required to drive a ship varies faster than the cube of the speed, but on the other hand, as the flap approaches closer to the skin of the ship, it is receiving water of proportionately lower velocity from the inner portion of the wake belt, which compensates for the fact that the power does not vary exactly as the cube. As a matter of fact, because of the reduction of the inlet area compared with the discharge area of the scoop, the static efficiency actually would improve with the reduced capacity.

If we are going to have movable parts, the movable lip will always be many times as efficient as movable blades in a pump, since there is not the loss in a prime mover, the energy always being furnished at the efficiency of the ship's propeller. The movable lip is many times more rugged than the movable propeller blades and not so tricky.

Dr. Marcello Orlando * has shown the type of pump inlet generally used in Italy and France, reproduced in Figure 3, and comments that a strong outflow has been observed out of the after half of the opening, which is made large to give a free inlet for maneuvering, while the forward portion is designed to favor the flow when going ahead. Dr. Orlando has commented on the fact that this causes a considerable waste of power.

As his examination of Figure 3 has already suggested the thought to the reader of putting a movable flap in the intake to prevent water from coming out of the after portion thereof, the writer will be pardoned for indicating in Figure 4 how this may be done.

Since we have a pin joint for the flap, we may just as well add the movable scoop lip, as in Figure 5, and, since there is now no back flow out of the rear portion of the entrance, we may, to advantage, omit the lip (of Figure 3) and round this with a large radius to give a free inlet while maneuvering.

* Machinery Superintendent O. T. O. Leghorn. A discussion of scoops in the Annals of the Italian National Tanks, Vol. 6, Rome, 1936.

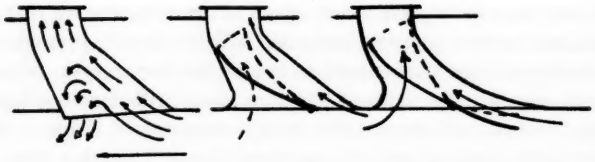


FIGURE 3. FIGURE 4. FIGURE 5

If we now started with a new design, we would arrive at something like the inlet shown in Figure 6.

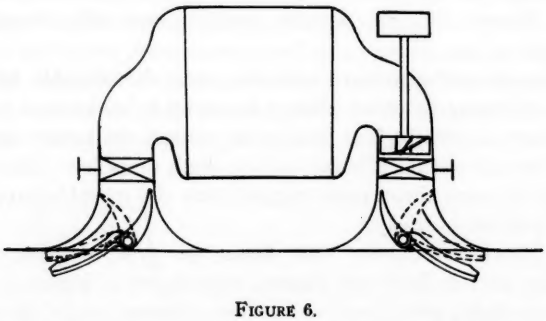


FIGURE 6.

While it is believed that experience on several ships having the standby propeller pump in the scoop discharge has shown that there is no ill effect in the form of deterioration of the water boxes, tube sheets or tubes as a result of the pump either built into or in close proximity to these parts, it may be pointed out that with the free inlet and discharge for water under all conditions in the arrangement shown in Figure 6 the propeller type standby pump may be put in the overboard discharge, thus making possible a much smoother delivery of water to the tubes than in any previous arrangement, since the water is forced through the tubes because of the reduction of pressure in the after water box. With ample passages and the water velocity through the tubes as at present limited to less than 10 feet, the suction at the pump under operating conditions is not sufficient to cause any difficulty. It may be pointed out that the suggestion of the pump located in the over-

board discharge was first made in connection with the underneath condenser arrangement presented to the Navy Department in 1927.

One of the objections raised to the movable lip scoop was that it would be dangerous in shallow water or in docking because of the danger of getting caught. However, danger with the arrangement of the by-pass check is not a matter of fact, since in shallow water or unknown waters the ship would be moving slowly and the lip can be moved out of danger, the water being taken in through the opening controlled by the check valve. The inlet lips and checks can be tied together so that they operate in unison.

In actually constructing a divergent scoop with movable lip automatically controlled, the operating piston would not be direct acting as shown, but would be controlled through a spring-resisted relay so that the lip would be held positively in both directions of motion by hydraulic pressure, the spring closing the flap at low speed or in case of damage to the operating mechanism. The type of divergent scoop shown in Figure 6 can be designed to have a cavitating speed of not less than 50 knots, a capacity of 35,000 to 40,000 gallons per minute and still occupy a vertical height of not over 30 inches from the skin of the ship.

In regard to the reference to predicting scoop performance, attention may be called to the fact that some dozen ships equipped with divergent scoops have all performed according to predictions within the limit of accuracy which would be expected in the estimated performance of a centrifugal pump. Previously, however, changes were made arbitrarily on scoops a number of times, believing that insufficient water was being obtained, whereas actually the lack of performance was entirely due to the condenser design.

Doctor Marcello Orlando has suggested that the performance of the circulating system could be improved by employing divergent scoops both on the inlet and discharge, using a circulating pump to overcome the pipe friction, condenser water box and tube resistances. Thus, if we assume that the inlet and the outlet are of such an area that when multiplied by the velocity of approach it gives the desired quantity of water, it is obvious that neglecting the frictional losses (provided by the pump) the work done in driving the ship by the overboard discharge would exactly equal the work expended on the circulating water by the scoop, hence,

the energy required to pick up and discharge the water from the ship would become zero, the work of forcing the water through the resistances of the system being provided by the pump as previously mentioned. For actual practical application, however, Dr. Orlando suggests a discharge velocity of approximately one-half the ship's velocity. The only disagreement which the writer has with Dr. Orlando is that if we use a discharge velocity of only one-half the ship's velocity, it will be unnecessary to run the circulating pump, since one-half the velocity corresponds to only one-quarter of the energy available at the scoop inlet and three-quarters of the energy available at the scoop inlet should be more than sufficient to overcome the resistance of the circulating pump, pipe friction and tube resistances. The efficiency of such a system will be extremely high.

In Figure 6 is shown a practical arrangement for designing a scoop system of the maximum possible efficiency with the safety required for naval vessels. In this system scoops with movable flaps are employed on the inlet and discharge, and in order to overcome the difficulty of providing the circulating pump with a free suction and discharge for maneuvering purposes, a check flap is incorporated in the inlet and overboard discharge scoops for permitting free entrance and exit of the water during maneuvering and when going astern, with the additional advantage that the additional passages may be so formed that when running astern they produce a considerable scoop and discharge action similar to ahead so that the efficiency of the system is substantially equal in ahead and astern operation and has no disadvantages when standing still. Under the latter condition water may enter through the scoop in whatever position it may happen to be and also through the check controlled inlet which gives free flow to the pump.

Since Mr. Kroon has shown that the power is the same regardless of whether the water is taken from the skin or still water, it is obvious that water of the highest velocity available should always be used, and the scoop entrance designed to fulfill this requirement.

It should be pointed out that Dr. Orlando's suggestion of discharging the water backward with the velocity of approach is only strictly effective if there is no wake belt, and it must not be forgotten that when water is discharged from the hull under water a

layer of water which has been accelerated to the velocity of the ship is displaced, and the rearwardly discharged water is in the end brought to the same velocity as that of the water displaced. Thus, the reaction of the discharged jet is partly if not wholly utilized in replacing the displaced wake belt. It is for this reason that a velocity of half that of approach is suggested as about the best compromise, unless it were considered permissible to discharge the water in air so that it landed beyond the wake belt. In this case practically the theoretical ideal would be obtained. If the water were discharged rearwardly with the ship's velocity, there would not be the splash which occurs when a 90-degree or low-velocity discharge comes out of water, since the jet would land with no relative velocity.

Prof. Weske's suggestion to use a velocity of six-tenths of the velocity of approach in the inlet for a pump is not practicable, since the velocity would be so high that the pump would not be capable of furnishing sufficient water when going astern and maneuvering. If a separate inlet is furnished for the pump, then the full velocity of approach should be used and a divergent tube employed to convert the velocity into pressure, thus reducing the velocity to that suitable for entrance to the water box.

Mr. H. Emmons,* using a different method of attack, has suggested the following way of arriving at the power absorbed by the circulating water.

To simplify the problem, it is assumed that the water taken in by the scoop is *stored on board.*

A ship S moving with a velocity V_o feet per second and storing water on board at a rate of V cubic feet per second is shown in Figure 9.

The propellers must overcome an additional resistance F, which is equal and opposite to that exerted by the water on the ship and represents an increase of power

$$P = F V_o \qquad (1)$$

To calculate F, consider a portion of the ocean within the imaginary frame A B C D moving with the ship. The frame is of such a contour that an observer on the ship would see water enter-

* Westinghouse Electric and Manufacturing Company.

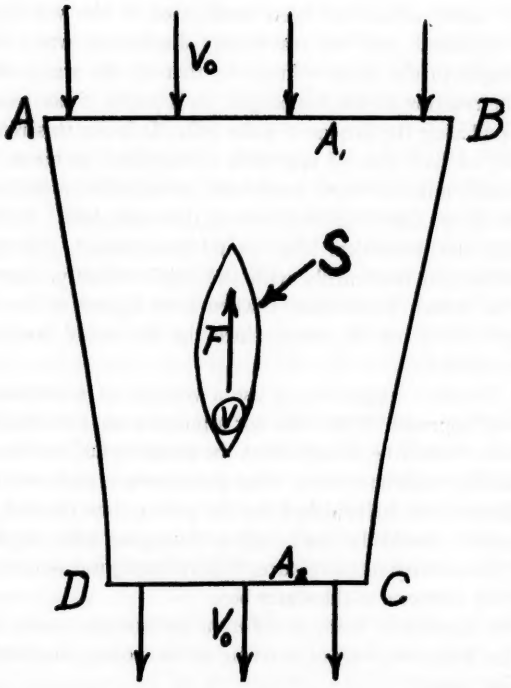


FIGURE 9.

ing through AB and out through DC at a rate V_o, but *none* would cross the sides AD and BC. Thus the area A_2 is less than A_1 because of the water withdrawn and stored on the ship.

We now have the momentum equation in the direction of V_o

$$F = \rho A_1 V_o^2 - \rho A_2 V_o^2 = \rho V_o^2 (A_1 - A_2) \qquad (2)$$

where ρ is the density of sea water.

Conservation of mass requires that

$$\rho A_1 V_o - \rho A_2 V_o = \rho V \text{ or } A_1 - A_2 = \frac{V}{V_o} \qquad (3)$$

Putting (3) into (2) gives the resistance F encountered by the ship

$$F = \rho V V_o \qquad (4)$$

Combining this with equation (1) gives

$$P = \rho V V_o^2 = 2 (\tfrac{1}{2} \rho V V_o^2) = 2 (\tfrac{1}{2} m V_o^2) \qquad (5)$$

Thus, to take water on board requires *twice* the kinetic energy of the water relative to the ship, and from the analysis it is evident that it is *independent* of the method by which it is taken on board.

It is proper to inquire of the whereabouts of this energy. $\frac{1}{2} \rho V V_o^2$ is obviously due to its motion with the ship. The other $\frac{1}{2} \rho V V_o^2$ may be available as static and kinetic pressure head to overcome various frictional and other resistances within or without the ship.

Particularly it should be noted that the question of the wake belt is not involved at all, the *only fact* which counts is that the water *enters* the ship no matter *how*.

In his article "Some Notes on E.H.P. Calculations," JOURNAL A.S.N.E., November, 1933, the writer made the error of stating the energy expended in picking up the circulating water was proportional to $W V^2/2g$. This should have been $W V^2/g$, consequently all the denominators in the various equations should be divided by 2. The writer made the same errors in deducing the value of the factor F, consequently the factor F should have been $\frac{1}{2}$ the value given. Thus the total result is not changed. However, if applying the factor F with the correct formula, the values of F as given should be divided by 2.

JOHN R. WESKE—Reply to comments on "Investigation of the Action of Condenser Scoops Based Upon Model Tests."

In his discussion Mr. Schmidt brings his experience with actual scoop installations and his familiarity with the wider implications of operating economy and requirements of marine installations to bear on the condenser scoop problem. It must be kept in mind, however, that the author in his investigations attempted to treat the scoop primarily as an energy problem. In particular, he had no intention of taking sides on the issue: pumps vs. self-actuating

scoop. The presence of an auxiliary fan in the test setup, which was necessary to run the scoop at normal capacity, can hardly be construed as such.

The author agrees with Mr. Schmidt that a hydrodynamic problem like that of the scoop requires rigor and attention to details in experimental analysis. The nature of the flow in the boundary layer, occurrence of flow separation, and mixing processes between streams of different total energy lead to regions of high vorticity and effect considerable deviations of actual characteristics from those of the frictionless fluid. But he fails to see why this should justify branding the classical approach by way of the ideal fluid as purely hypothetical, particularly since Mr. Schmidt himself has made generous use of this approach on previous occasions.[1]

The discussion calls for further comments in regard to accuracy and reliability of the results and of wind tunnel work in general.

The working section of the wind tunnel has a square cross section of 3 by 3 feet, bounded by a ceiling—in which the scoop was installed—and a floor. The sides are open to the atmosphere. Thus, the wind tunnel is basically of the open gap type and consequently, the static pressure of undisturbed flow is atmospheric. The quantity of air removed or added never exceeded 1 per cent of the wind tunnel stream. Elimination of errors of measurement from this source are a matter of routine in wind tunnel work.

After initial difficulties had been overcome the method of boundary layer control by means of removal of air through an upstream slot worked satisfactorily. It was possible to obtain velocity traverses typical of a turbulent boundary layer of the desired momentum thickness. These traverses were taken with a small pitot tube (hypodermic needle) before each test. A boundary layer of ¼ inch thickness on the model corresponds to one of 3 inches on the ship, thus comparing favorably with measured boundary layers.

In most cases it was possible to read pressures with an accuracy of within 2/100 inch H_2O. Since the auxiliary fan ordinarily is used for even more exacting work, sufficient precaution was

[1] F. H. Schmidt, *Theoretical and Experimental Study of Condenser Scoops*, "JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS," Feb., 1930.

taken against errors in quantity measurements caused by rotation of the stream.

Mr. Schmidt takes exception to the fact that the total pressure produced by the scoop decreases as the quantity approaches normal capacity, $\frac{V_1}{V_0} = 1$. It appears, however, that his reasoning, which leads him to expect a value but slightly smaller than the velocity pressure of undisturbed flow past the scoop, holds only if the velocity across the scoop area is constant. This is undoubtedly not the case, as he has pointed out previously.[2] A scoop is more aptly compared to an elbow than to a straight piece of pipe, and it is well known that the loss of head of the former is many times that of the latter. The author's pressure measurements of intake scoops compare very closely with data hitherto unpublished of a similar intake obtained in connection with aircraft development work by one of the leading research institutions.

Lack of time prevents the author from preparing for publication at this time experimental data in support of his statement in regard to the effect of aspect ratio of intakes upon their efficiency. Recent trend in the design of intakes on aircraft has been toward larger aspect ratios since these were found to give higher efficiencies. That the efficiency of discharges increases with aspect ratio can be seen by comparing the characteristics of a discharge slot published elsewhere[3] with those of the circular discharge presented in the paper under discussion.

The formation of a forward force and of a back pressure for a discharge normal to a plate was demonstrated and explained in greater detail at the Fifth International Congress of Applied Mechanics.[4] According to the model tests the percentage of energy recovered from the discharge through this propulsive force is such that it should not be neglected (for the conditions of the numerical example given below, 275 HP, or 35 per cent, are recovered of 800 HP discarded overboard with the discharge).

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[2] Figs. 7 and 8, pp. 9 and 10 reference (1).

[3] J. R. Weske, *Reduction of Skin Friction of a Flat Plate Through Boundary Layer Control,* "Journal of the Aeronautical Sciences," May, 1939, p. 289.

[4] Applied Mechanics (*proceedings of the 5th International Congress of Applied Mechanics,* 1938, p. 464). Discussion of the paper by A. Métral, by J. R. Weske.

Certain observations made investigations of the effect of the wake upon skin friction along the ship's hull appear desirable. These investigations disclosed that for the case of a flat plate there is a measurable reduction of drag due to the blanketing effect of the discharged air moving past the plate at lower speed.[3] The information available in regard to the matter at the moment is not adequate, however to lend support to a prediction that a similar reduction of skin friction will take place on a curved surface, such as that of a ship's hull.

In his decision not to include in his paper overboard discharges with lips, the author was led by considerations of efficiency and not of capacity. He assumed that, in case the lip was designed to follow exactly the stream surface separating the jet from the main stream, the power required would exactly equal that of the flush discharge. In all other cases the lip would cause additional disturbance of the flow and therefore require more power. If increase of pressure drop is at a premium compared with efficiency, however, or if power is considered at the source and not at the point of application the lip may have distinct merits.

The suitability for practical application of the manner of presentation of test results by means of non-dimensional quantities, which Mr. Schmidt questions, can be demonstrated best by a numerical example. Since for obvious reasons scoop models used in the tests do not correspond to latest design practice, the results are not expected to apply numerically to modern installations. They should, however, give a good indication of the order of magnitude of the quantities involved in scoops of present-day design.

NUMERICAL EXAMPLE OF SCOOP CALCULATION.

Data given: Speed of ship 40 KN.
Quantity of cooling water 30,000 GPM.
Area of intake and discharge 3.3 sq. ft.

a. Performance of intake.

Nominal velocity through intake $V_1 = 20$ ft/sec
Velocity of ship $V_o = 67.6$ ft/sec

$$\text{Velocity ratio } \frac{V_1}{V_o} = 0.3$$

from Figure 5 of the paper: $C_D = 0.66$
$C_p = 0.44$
$e_t = 0.43$

Drag of intake $D = C_D A \rho \frac{V_o^2}{2} = 9850$ lb.

Static head produced by scoop $h_s = C_p \frac{V_o^2}{2g}$
$= 31.2$ ft. H_2O

Velocity head produced by scoop $h_v = \ldots. 6.4$ ft. H_2O
Total head produced by scoop $h_t = 37.6$ ft. H_2O

Horsepower requirements of intake $HP_1 = \frac{DV_o}{550} = 1210$ HP.

b. Performance of discharge.

Velocity ratio $\frac{V_2}{V_o} = 0.3$

for discharge inclined at 30 degrees aft, Figure 9 of paper:

$C_D = -0.16$
$C_p = 0.22$
$e_2 = 0.42$

Negative drag (propulsive force) $D = C_D A \rho \frac{V_o^2}{2}$
$= 2420$ lb.

Back pressure at discharge $h_s = C_p \frac{V_o^2}{2g}$
$= 15.6$ ft. H_2O

Velocity head at discharge $h_v = \frac{V_2^2}{2g}$
$= 6.4$ ft. H_2O

Total head at discharge (with respect to ship) $h_t \ldots$
$= 22.0$ ft. H_2O
Latent velocity head (from motion of ship relative to water) $h'_v = \frac{V_o^2}{2g} = 71.0$ ft. H_2O

Gross total head at discharge (with respect to sea)

$$h'_t = 93.0 \text{ ft. } H_2O$$

Energy in water discharged overboard

$$HP_2 = \frac{\text{Weight of } H_2O/\text{sec} \times h'_t}{550} = 720 \text{ HP.}$$

Energy recovered through propulsive force

$$HP_r = \frac{DV_o}{550} = 300 \text{ HP} = 42 \text{ per cent of } HP_2.$$

Neglecting losses in the connecting piping the pressure drop across the condenser is

$$\triangle h = 31.2 - 15.6 = 15.6 \text{ ft. } H_2O.$$

In the case of a discharge normal to, and flush with, the hull the pressure drop across the condenser would be

$$\triangle h = 5.6 \text{ ft. } H_2O$$

This would necessitate the application of a lip to the overboard discharge to produce a pressure drop adequate for proper operation of the condenser.

Scoop installations designed to operate at a velocity ratio $\frac{V_1}{V_o}$ of approximately 0.3 were found by the author to produce maximum static pressure.

ASSOCIATION NOTES.

CHANGE IN COUNCIL.

Commander E. L. Cochrane (CC) U. S. Navy resigned as Member of the Council of the Society because of detachment from duty in Washington. Captain Julius A. Furer (CC) U. S. Navy, was appointed to membership to fill the unexpired term of Commander Cochrane.

MEMBERSHIP.

The following have joined the Society since the publication of the May, 1939 issue of the JOURNAL:

NAVAL.

Buerkle, Elmer C., Lieutenant, U. S. Navy.

Byrd, John H., Lieut. Commander, U. S. Coast Guard.

Carmick, Edward S., Lieutenant, U. S. Navy.

Casey, Kendall, Ensign, U. S. Navy, Retired, 701 Springs Road, Vallejo, Calif.

Crampton, D. K., Lieut. Commander, U. S. N. R., Director of Research, Chase Brass and Copper Co., Waterbury, Conn.

Croft, Huber O., Lieut. U. S. N. R., College of Engineering, University of Iowa, Iowa City, Iowa.

Dick, George William, Lieutenant, U. S. Coast Guard.

Gerde, C. S., Ensign, U. S. Coast Guard.

Grove, Marvin N., Lieutenant, U. S. Navy, Retired, Grove Regulator Co., 1729 Poplar St., Oakland, Calif.

Lindauer, G. C., Lieutenant, U. S. Coast Guard.

Palmer, William G., U. S. Navy.

Smenton, John J., Lieutenant, U. S. Coast Guard.

Sutherland, Alan B., Jr., Lieutenant, U. S. Navy.

Usina, M. N., Captain, U. S. Coast Guard, Retired, 57 Lake View Avenue, Newton Centre, Mass.

Weakley, Charles E., Lieutenant, U. S. Navy.
Weller, D. W., Lieutenant, U. S. Coast Guard.

CIVIL.

Day, Charles L., Owens-Corning Fiberglas Corporation, 718 Fifth Avenue, New York, N. Y.
Gould, E. H., Jr., 927½ Carolina St., Vallejo, Calif.

ASSOCIATE.

Mikolic, Michael, First Asst. Engineer, MS "Benson Ford", 4721 Springwells Avenue, Detroit, Mich.

BOOK REVIEW.

PRINCIPLES OF NAVAL ARCHITECTURE (TWO VOLUMES). By The Society of Naval Architects and Marine Engineers. $11.00. Reviewed by Rear Admiral S. M. Robinson, U. S. Navy.

This is a comprehensive book on the subject of Naval Architecture and has been written from a wholly new point of view. It is in condensed form, the whole book requiring only 284 pages in the first volume and 232 pages in the second volume. However, the greatest care has been used in condensing, so that none of the essentials are omitted; it has been possible to do this only because each chapter has been written by an expert.

No attempt is made to cover any of the practical side of shipbuilding. The volume is purely a theoretical exposition of the art of designing a ship and of the performance of the ship in the water after it has been built. It is believed that this is an excellent subdivision of the subject matter. It is expected that the Society will bring out a book on practical Naval Architecture in the near future to accompany the present splendid volume.

ERRATA.

August, 1939, DISCUSSIONS AND COMMENTS.

Page 463, line 3 should read :

$$\text{approaching } \frac{\delta}{g} \cdot V \cdot v_o^2 \text{ for the limit } \Delta x = o$$

in place of

$$\text{approaching } \frac{\delta}{g} \cdot V \cdot v_o^2 \text{ for the limit } \Delta x = o$$